EP 30311 (1)

(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: 0 392 500 A2

(12) **EUROPEAN PATENT APPLICATION**

(21) Application number: 90106941.9

(51) Int. Cl.[5]: G11B 19/20, H02K 7/04

(22) Date of filing: 11.04.90

(30) Priority: 12.04.89 JP 92161/89
12.07.89 JP 179647/89
08.08.89 JP 205077/89
30.08.89 JP 223679/89

(43) Date of publication of application:
17.10.90 Bulletin 90/42

(84) Designated Contracting States:
AT BE CH DE ES FR GB IT LI NL SE

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(54) **Spindle motor.**

(57) A spindle motor comprising a stator including a support shaft positioned on a base, a cap-shaped rotor rotatably and concentrically disposed around the support shaft, thrust and radial bearings disposed between the stator and the rotor, a stator coil secured to the stator, and a rotor magnet member secured to the rotor in opposing relation to the stator coil. The thrust and radial bearings are hydrodynamic bearings. A movable piece that constitutes a part of the thrust bearing is secured to the lower end of a cylindrical portion of the rotor and extended outwardly or inwardly from the cylindrical portion, or is disposed above the rotor. The stator coil is secured to the outer peripheral portion of the support shaft above or below the radial bearing. The rotor magnet member is secured to the inner peripheral surface or ceiling of the rotor. Either a radial or axial gap is provided between the stator coil and the rotor magnet member.

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SPINDLE MOTOR

The present invention relates to a spindle motor capable of rotating at high speed, which employs hydrodynamic bearings as radial and thrust bearings. More particularly, the present invention relates to a spindle motor which is designed to rotate with minimal vibrations irrespective of the position of the motor when used and hence is suitable for a hard disk driver (hereinafter referred to as simply "HDD").

With the achievement of HHDs with a high storage capacity and low power consumption, demand has been made for improvements in the performance of spindle motors which are used to drive them so as to be even more suitable therefor.

Fig. 16 schematically shows a conventional spindle motor which is actually used in an HDD, and Fig. 15 is a partially sectioned elevational view of the conventional spindle motor. The spindle motor 20 has a shaft support cylinder 22 in the center of a mount 21. A stator coil 23 which comprises a plurality of electromagnetic coils is secured to the outer periphery of the shaft support cylinder 22. A rotary shaft 25 is rotatably supported by the inner periphery of the shaft support cylinder 22 by ball bearings 24. The rotary shaft 25 has a support member 27 secured to the upper end thereof, the support member 27 being arranged such that hard disks 30 are fixedly mounted on the outer peripheral surface thereof. The support member 27 has a plurality of rotor magnet members 28 secured to the inner peripheral surface in opposing relation to the stator coil 23.

In the above-described spindle motor employing ball bearings, the magnitude of vibrations of the spindle motor depends on the internal clearances of the ball bearings. The magnitude of vibrations in the radial direction is substantially equal to the radial internal clearance of the ball bearings. Similarly, the magnitude of vibrations in the thrust direction is substantially equal to the thrust internal clearance of the ball bearings. Measures have been taken to reduce these internal clearances, for example, by preloading the ball bearings. However, no satisfactory internal clearance value has heretofore been obtained, i.e., it has been only possible to achieve 0.5 microns or so in terms of the non-repeated component of the runout in the radial direction. In addition, preloading of ball bearings results in an increase in the required torque of the motor instead and hence retrogresses to the desirous lowering in the power consumption of the HHD. Accordingly, as long as ball bearings such as those described above are used, it is in principle virtually impossible to further reduce the vibrations of the spindle motor.

In addition, it is necessary to use small-sized ball bearings for the above-described spindle motor, which involves the problem that the HHD is not shock resistant and is therefore inferior in durability. With regard to the achievement of high speed, i.e., 3,600 rpm → 5,400 → rpm 6,400 rpm, the prior art suffers from the problem of how to minimize the wear of the bearings. Further, since the prior art uses a lubricating oil (grease), the degree of cleanliness is low.

In case of a spindle motor whose bearings are only replaced by hydrodynamic bearings, it still suffers from the disadvantage that rotational vibration increases when they are used in a horizontal position. In addition, since the clearance between a movable piece and a fixed piece of a radial bearing is of the order of microns, when two discrete radial bearings are used, it is difficult to align them concentrically. When two discrete thrust bearings are used, it is difficult to adjust the relative position between the thrust bearings. In addition, since the thrust collar, that is, movable piece, of a thrust bearing is produced so that the parallelism is within several microns, it is necessary to hold down the parallelism to about 1 micron when it is assembled, which is very difficult. In a radial gap type spindle motor wherein a rotor magnet member is disposed around the outer periphery of the stator coil with a radial gap provided therebetween, moment is generated due to the imbalance of radial magnetic force acting between the stator coil and the rotor magnet member, causing the axis of the rotor to be inclined with respect to the support shaft, which results in an increase in the starting torque because of local contact of the dynamic pressure surfaces. In rotation, unstable radial magnetic force causes whirling of the shaft and therefore makes it impossible to obtain a satisfactory operating condition.

It is an object of the present invention to provide a spindle motor which employs hydrodynamic bearings to improve the high-speed rotating performance and minimize the vibration irrespective of the position of the motor when used and which is therefore suitable for a high-recording capacity HDD.

The present invention provides a spindle motor which employs hydrodynamic bearings to improve the durability, clean operation and high-speed rotating performance and minimize vibrations when rotating irrespective of the position of the motor when used and which is therefore suitable for a high-recording capacity HDD.

The spindle motor of the present invention comprises a stator including a support shaft stood

on a base, a cap-shaped rotor rotatably and concentrically disposed around the support shaft, thrust and radial bearings disposed between the stator and the rotor, a stator coil secured to the stator, and a rotor magnet member secured to the rotor in opposing relation to the stator coil. The thrust and radial bearings are hydrodynamic bearings.

The spindle motor of the present invention may be arranged such that a movable piece which constitutes a part of the thrust bearing is secured to the lower end of a cylindrical portion of the rotor and extended outwardly from the cylindrical portion, a fixed piece which constitutes another part of the thrust bearing is fixed to the base in opposing relation to the movable piece, the stator coil is secured to the outer peripheral portion of the support shaft above the radial bearing, and the rotor magnet member is secured to the inner peripheral surface of the rotor so that a radial gap is provided between the stator coil and the rotor magnet member.

The arrangement may also be such that the stator coil is secured to the outer peripheral portion of the support shaft above the radial bearing and the rotor magnet member is secured to the ceiling of the rotor so that an axial gap is provided between the stator coil and the rotor magnet member.

The present invention may be arranged such that the movable piece of the thrust bearing is secured to the lower end of the cylindrical portion of the rotor and extended inwardly from the cylindrical portion, the fixed piece of the thrust bearing is secured to the base in opposing relation to the movable piece. In this case, both the stator coil and the rotor magnet member may be disposed above the radial bearing, the rotor magnet member being secured to the ceiling of the rotor so that an axial gap is provided between the stator coil and the rotor magnet member.

The present invention may be arranged such that the movable piece of the thrust bearing is secured to the lower end of the cylindrical portion of the rotor and extended outwardly from the cylindrical portion, the fixed piece of the thrust bearing is secured to the base in opposing relation to the movable piece, and the rotor magnet member is disposed at a position which is inward of the movable piece of the thrust bearing. In this case, the stator coil may be secured to the base so that an axial gap is provided between the stator coil and the rotor magnet member. The stator coil may be secured to the lower part of the support shaft so that a radial gap is provided between the stator coil and the rotor magnet member.

The present invention may be arranged such that the thrust bearing is disposed above the radial bearing, and both the stator coil and the rotor magnet member are disposed below the radial bearing. In this case, the stator coil may be secured to the base so that an axial gap is provided between the stator coil and the rotor magnet member.

In the present invention, the thrust bearing may be preloaded by magnetic force which acts counter to the thrust dynamic pressure. Accordingly, in a spindle motor wherein a radial gap is provided between the stator coil and the rotor magnet member, the thrust bearing is preloaded in the counter direction to the thrust dynamic pressure by offsetting the axial magnetic force center of the rotor magnet member from the axial magnetic force center of the stator coil by a predetermined amount in a counter direction to the thrust dynamic pressure. The axial magnetic force centers of the rotor magnet member and the stator coil are defined as those that, when there is an axial distance between them, an axial magnetic force is generated between them to reduce the distance.

In the present invention, the radial bearing is disposed so as to bear the rotor over a predetermined range including the center of gravity of the rotor.

In the present invention, the fixed piece of the thrust bearing is secured to the base through a resilient pad, for example, silicone rubber. The movable piece of the thrust bearing is secured to the rotor through a resilient pad, for example, silicone rubber.

In the present invention, the opposing annular sliding surfaces of the fixed and movable pieces of the thrust bearing are made of a ceramic material, for example, silicon carbide, alumina, etc., and either of the annular sliding surfaces has spiral grooves for generating dynamic pressure.

In the present invention, the opposing cylindrical sliding surfaces of the fixed and movable pieces of the radial bearing are made of a ceramic material, for example, silicon carbide, alumina, etc., and either of the cylindrical sliding surfaces has herringbone-shaped grooves for generating dynamic pressure.

In the present invention, the movable piece of the radial bearing, the rotor and the movable piece of the thrust bearing may be arranged in an integral structure. In this case, the moving piece of the radial bearing and/or the moving piece of the thrust bearing in the integral structure is coated with a kind of material different from that of groundwork thereof, or the groundwork thereof is surface treated.

In the present invention, the fixed piece of the radial bearing, the support shaft, the fixed piece of the thrust bearing and the base may be arranged in an integral structure. In this case, the fixed piece of the radial bearing and/or the fixed piece of the

thrust bearing in the integral structure is coated with a kind of material different from that of groundwork thereof, or the groundwork thereof is surface treated.

In the present invention, the rotor has a support member adapted to hold hard disks on the outer peripheral surface thereof.

In the spindle motor of the present invention, the support shaft may be extended through a through-hole provided in the upper end portion of the rotor and the distal end of the support shaft may be secured to a stationary part, thus the support shaft being loosely fitted in the through-hole.

In the spindle motor of the present invention, the upper end of the rotor may be closed and not pierced with the support shaft.

Figs. 1 to 12 are sectional views respectively showing the structures of various embodiments of the spindle motor according to the present invention;

Fig. 13 schematically shows dynamic pressure generating grooves formed in a radial bearing member;

Fig. 14 schematically shows dynamic pressure generating grooves formed in a thrust bearing member;

Fig. 15 is a partially sectioned elevational view of a conventional spindle motor for an HDD; and

Fig. 16 is a perspective view of the conventional spindle motor when actually used in an HDD.

Figs. 1 to 12 are sectional views respectively showing the structures of various embodiments of the spindle motor according to the present invention, in which the same reference numerals denote the same or corresponding portions.

Referring first to Fig. 1, which shows a first embodiment of the spindle motor according to the present invention, reference numeral 1 denotes a base, which has a support shaft 2 stood on the central portion thereof. A fixed piece 4b which constitutes a part of a radial bearing 4 is concentrically secured to the outer periphery of the support shaft 2. A stator coil 5 is secured to the support shaft 2 above the radial bearing 4. The stator coil 5 includes a plurality of electromagnetic coils equally spaced in the circumferential direction. A rotor 6 which serves as a hard disk supporting member has a cap-shaped configuration. The rotor 6 has a through-hole 16 provided in the upper end portion thereof. The upper end portion of the support shaft 2 extends through the through-hole 16. The distal end of the support shaft 2 may be secured to a stationary part (not shown). The upper end portion of the support shaft 2 is loosely fitted in the through-hole 16.

The rotor 6 has an annular bearing member 7 secured to the lower end portion thereof, the bearing member 7 having an L-shaped cross-sectional configuration. A rotor magnet member 8 is secured to the rotor 6 above the bearing member 7. The rotor magnet member 8 includes a plurality of magnets or iron cores equally spaced in the circumferential direction. The rotor magnet member 8 and the stator coil 5 face each other across a radial gap, thus constituting a drive part of the spindle motor.

The lower end portion of the bearing member 7 defines a movable piece 3a which constitutes a part of a thrust bearing 3. The movable piece 3a faces a fixed piece 3b which is secured to the base 1 to constitute another part of the thrust bearing 3. The inner peripheral surface of the bearing member 7 defines a movable piece 4a of the radial bearing 4. The movable piece 4a faces the fixed piece 4b of the radial bearing 4 that is secured to the support shaft 2.

The opposing cylindrical sliding surfaces of the movable and fixed pieces 4a and 4b of the radial bearing 4 are made of a ceramic material, for example, silicon carbide, alumina, etc., and either of the sliding surfaces has herringbone-shaped grooves C_1 for generating dynamic pressure, such as those shown in Fig. 13, the other sliding surface being smoothed.

The opposing annular sliding surfaces of the movable and fixed pieces 3a and 3b of the thrust bearing 3 are made of a ceramic material, for example, silicon carbide, alumina, etc., and either of the annular sliding surfaces has spiral grooves C_2 for generating dynamic pressure, such as those shown in Fig. 14, the other sliding surface being smoothed. The sliding surfaces may be made of a member coated with a kind of material different from the groundwork thereof, or may be made of a member having a treated surface of degenerated groundwork thereof, instead of the ceramic material.

The rotor 6 is arranged such that a plurality of hard disks (not shown) can be mounted on the outer peripheral surface of the upper part of the rotor 6 through a spacer.

In the spindle motor shown in Fig. 1, as the electromagnetic coils that constitute the stator coil 5 are sequentially supplied with an electric current, the rotor 6 having the rotor magnet member 8 secured thereto begins to rotate and consequently a hydrodynamic pressure is generated between the opposing annular sliding surfaces of the movable and fixed pieces 3a and 3b of the thrust bearing 3, thus forming a hydrodynamic thrust bearing. Similarly, a hydrodynamic pressure is generated between the opposing cylindrical sliding surfaces of the movable and fixed pieces 4a and 4b of the radial bearing 4, thus forming a hydrodynamic ra-

dial bearing.

Since the rotor 6 is supported in such a manner that the lower end and inner peripheral surfaces of the bearing member 7 are not in solid contact with the fixed piece 3b of the thrust bearing 3 and the fixed piece 4b of the radial bearing 4, the spindle motor is capable of smoothly rotating at high speed. Accordingly, the spindle motor of the present invention is free from the problem of friction and vibration in contrast to the prior art that employs ball bearings to support the rotor.

Some or all of the following elements, i.e., the base 1, the support shaft 2, the fixed piece 4b of the radial bearing 4 and the fixed piece 3b of the thrust bearing 3, may be formed in an integral structure from the same constituent material. The movable piece 4a of the radial bearing 4 and the movable piece 3a of the thrust bearing 3 are formed in an integral structure of the bearing member 7 having an L-shaped cross section. The rotor 6 and the bearing member 7 may be formed in an integral structure from the same constituent material.

Since the thrust bearing 3 is disposed outside the motor driving part that comprises the rotor magnet member 8 and the stator coil 5, if the dynamic pressure generating grooves C_2 are formed so that the dynamic pressure generated will act inwardly, air is sucked in from the outer peripheral side of the thrust bearing 3 and no air flows outward from the lower end portion of the rotor 6. No dust will therefore be scattered outwardly from the rotor magnet member 8 and the stator coil 5. Accordingly, the spindle motor of the present invention is suitable for use in an environment where dust must be kept out. In an environment where there is no particular need to used dust out, the dynamic pressure generating grooves may be formed so that the dynamic pressure generated will act outwardly of the thrust bearing 3.

In the spindle motor of the first embodiment shown in Fig. 1, a radial gap is provided between the rotor magnet member 8 and the stator coil 5, and the magnetic force center of the axial length of the rotor magnet member 8 is a distance d offset from the magnetic force center of the axial length of the stator coil 5, so that a magnetic force acts so as to make the center of the rotor magnet member 8 coincident with the center of the stator coil 5, thus enabling the thrust bearing 3 to be preloaded. The magnitude of the preload can be set at a desired value by varying the distance d.

The spindle motor shown in Fig. 1 may be arranged in the form of a synchronous motor by comprising the rotor magnet member 8 of a group of rotor magnets, and may also be arranged in the form of an induction motor by comprising the rotor magnet member 8 of a group of rotor cores.

Since the bearing member 7 that constitutes the movable pieces 4a and 3a of the radial and thrust bearings 4 and 3 rotates relative to the fixed pieces 3b and 4b, without contacting the latter, through a fluid which is compressed when the bearing member 7 rotates, these members that constitute the radial and thrust bearing 4 and 3 may be made of any kind of material as long as it can be machined with a high degree of accuracy. Any of generally employed metallic materials and organic materials may be utilized. The point is that it is necessary to minimize the frictional resistance and wear of the bearing members at the time when the motor is started and rotating at low speed. The range of usable materials therefore depends upon the bearing structure adopted.

In the first embodiment shown in Fig. 1, the size of each of the radial and thrust bearings 4 and 3 is increased to reduce the surface pressure acting on the contact surfaces and the stator coil 5 is properly disposed to attain a structure which is free from any local contact. Accordingly, if the members that constitute the bearings are made of, for example, a stainless steel, and a thin coat of lubricant is applied to the contact surfaces, it is possible to maintain a stable performance for a long period of time. However, no lubricant or only minimal lubricant can be used in certain environments where the spindle motor is used. In such a case, it is preferable to employ a material which is superior in wear-resistant and sliding properties, particularly a ceramic material. Silicon carbide or alumina is particularly suitable for such an application.

Since the clearance between the movable and fixed pieces of each of the thrust and radial bearings 3 and 4 is of a small value in the order of microns, the thrust bearing 3 is preferably disposed exactly at right angles with respect to the radial bearing 4. It is, however, difficult to dispose it exactly at right angles because of the limitation on the degree of accuracy with which the bearings are produced. For this reason, a resilient pad 12 which is made of a resilient material is interposed between the fixed piece 3b of the thrust bearing 3 and the base 1 to absorb any error in the perpendicularity. Any flexible and durable material may be employed to form the resilient pad 12. However, silicone rubber is suitable from the viewpoint of both flexibility and durability.

In the first embodiment, the support shaft 2 is extended through the through-hole 16 provided in the upper end portion of the rotor 6 and the distal end of the support shaft 2 may be secured to a stationary part (not shown). In such a case, the support shaft 2 is supported at both ends by the base 1 and the stationary part and it is therefore possible to prevent deflection of the support shaft 2 which would otherwise be caused by the weight of

the load attached to the rotor 6 when the spindle motor is used in a horizontal position. The structure in which the support shaft 2 is supported at both ends is advantageous for use in a large-sized spindle motor.

The radial hydrodynamic bearing 4 that comprises the movable and fixed pieces 4a and 4b is sufficiently long to cover the range of from the lower end of the support shaft 2 to the center of gravity G of a rotary assembly comprising the rotor 6 and hard disks mounted thereon. As the length of the radial hydrodynamic bearing 4 increases, the effective working pressure range within which sufficient load carrying capacity is provided increases, and the radial vibration decreases. Further, since the radial hydrodynamic bearing 4 is not formed in a cantilever structure, the starting torque is minimized. Since the radial hydrodynamic bearing 4 is long and a large dynamic pressure is therefore generated, the movable and fixed pieces 4a and 4b of the radial bearing 4 is not required to be machined to any particularly high degree of accuracy.

When the spindle motor is used in a vertical position, the range of preload applied in the thrust direction by the magnetic force from the rotor magnet member 8 depends on the dynamic pressure generated between the movable and fixed pieces 3a and 3b of the thrust bearing 3, the weight of the rotor 6 that is applied to the thrust bearing 3 and the machining accuracy of the movable and fixed pieces 3a and 3b of the thrust bearing 3. However, it is, basically, only necessary to satisfy the following relationship:

$P < 100 \times S^2 - W$ (1)

wherein P: the preload [g] applied by the rotor magnet member 8; S: the area [cm^2] of the thrust bearing 3; and W: the weight [g] of the rotor 6. In the above relationship, $100 \times S$ is the dynamic pressure [g/cm^2] required for the rotor to rotate without solid contact through a hydrodynamic bearing finished by an existing, economical finishing process.

In the first embodiment, since the thrust bearing 3 is disposed outside the lower end of a cylindrical portion of the rotor 6, that is, the driving part comprising the rotor magnet member 8 and the stator coil 5, the diameter of the thrust bearing 3 increases, and the rotor 6 is pulled by the above-described preload toward the thrust bearing 3 having a relatively large diameter. Accordingly, the radial deflection of the rotor 6 decreases, and stable rotation of the rotor 6 is achieved.

Fig. 2 is a sectional view showing the structure of a second embodiment of the spindle motor according to the present invention, which is similar to the first embodiment shown in Fig. 1 but different therefrom in the following point. In the second embodiment, the upper end of the rotor 6 is closed and not pierced by the support shaft 2. In addition, a member which corresponds to the annular bearing member 7 having an L-shaped cross-sectional configuration, shown in Fig. 1, is split into a radial bearing sleeve, that is, a movable piece 4a of a radial bearing, and a thrust bearing collar, that is, a movable piece 3a of a thrust bearing. The movable piece 4a of the radial bearing is secured to the inner peripheral surface of the rotor 6 in opposing relation to the fixed piece 4b of the radial bearing. The movable piece 3a of the thrust bearing is secured to the lower surface of a collar portion 17 formed at the lower end of the rotor 6, the movable piece 3a facing the fixed piece 3b of the thrust bearing.

When the spindle motor of the first or second embodiment is used in a horizontal position, if no preload is applied in the thrust direction by magnetic force, the weight of the rotor 6 is not applied to the thrust bearing 3. Accordingly, an inclination of the support shaft 2 with respect to the radial bearing causes local contact of the bearing members. When the thrust bearing 3 starts to generate a dynamic pressure in such a state, the rotor 6 is sprung out in the thrust direction and is therefore unable to rotate stably. It is therefore preferable to apply in advance a force to the thrust bearing 3 in a counter direction to the dynamic pressure generated from the thrust bearing 3, that is, to preload the thrust bearing 3, by previously offsetting the magnetic force center of the rotor magnet member 8, which constitutes the drive part of the spindle motor, from the axial magnetic force center of the stator coil 5 by a distance d in the direction in which the rotor magnet member 8 comes away from the annular thrust bearing 3.

Fig. 3 is a sectional view showing the structure of a third embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 2 but different therefrom in the following point. In this spindle motor, the rotor magnet member 8 is disposed on the ceiling of the rotor 6. The stator coil 5 comprises a plurality of electromagnetic coils which are secured at equal spacings to the outer periphery of the support shaft 2 above the fixed piece 4b of the radial bearing 4. Whereas the spindle motors shown in Figs. 1 and 2 are of the radial gap type, the spindle motor shown in Fig. 3 is of the thrust gap type.

In contrast to the radial gap type spindle motors, the thrust gap type spindle motor is free from the moment which would otherwise be generated due to the imbalance in radial magnetic force acting between the stator coil and the rotor magnet member, and the vibration of the motor when rotating is therefore reduced.

Fig. 4 shows the structure of a fourth embodi-

ment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 1 but different therefrom in the following point. In the spindle motor shown in Fig. 4, the base 1, the support shaft 2, the fixed piece 4b of the radial bearing and the fixed piece 3b of the thrust bearing are formed in an integral structure from the same constituent material. Similarly, the rotor 6, the movable piece 4a of the radial bearing and the movable piece 3a of the thrust bearing are formed in an integral structure from the same constituent material. The support shaft 2 is extended through a through-hole 16 provided in the upper end portion of the rotor 6, and the distal end of the support shaft 2 is secured to a stationary part 42. Either one or both of the moving and fixed pieces of the radial and thrust bearings in the fourth embodiment may be coated with thin film of a kind of material different from that of groundwork thereof, or may be provided with a treated surface layer of degenerated groundwork thereof. The thin film may be made, e.g. by way of physical vapor deposition or chemical vapor deposition, or by way of plating with a kind of material different from the groundwork thereof. The treated surface layer may be made, e.g. by way of oxidation, nitriding, or ion implantation with the groundwork of the pieces.

Fig. 5 shows the structure of a fifth embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 1 but different therefrom in the following point. In this embodiment, a thrust bearing 3 which has relatively small diameter is disposed below a radial bearing 4. A member which corresponds to the bearing member 7 having an L-shaped cross-sectional configuration, shown in Fig. 1, is split into a movable piece 4a of the radial bearing 4 and a movable piece 3a of the thrust bearing 3. The movable piece 4a of the radial bearing 4 is secured to the inner peripheral surface of the rotor 6 in opposing relation to a fixed piece 4b of the radial bearing 4 which is secured to the support shaft 2. The movable piece 3a of the thrust bearing 3 is secured to the inner peripheral surface of the lower end portion of the rotor 6 in opposing relation to a fixed piece 3b of the thrust bearing 3 which is secured to the base 1 through a resilient pad 12.

In the fifth embodiment, a motor driving part which comprises a stator coil 5 and a rotor magnet member 8 is disposed above the radial bearing 4. The stator coil 5 is secured to the outer periphery of the support shaft 2 above the radial bearing 4, and the rotor magnet member 8 is secured to the inner peripheral surface of the rotor 6 in opposing relation to the stator coil 5.

Fig. 6 shows the structure of a sixth embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 5 but different therefrom in the following point. In the sixth embodiment, the movable pieces 3a and 4a of the thrust and radial bearings 3 and 4 are joined together in the form of an integral bearing member 7. This structure facilitates assembly of the bearings.

Fig. 7 shows the structure of a seventh embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 5 but different therefrom in the following point. In the seventh embodiment, the motor drive part comprises a rotor magnet member 8 which is secured to the ceiling of the rotor 6 and a stator coil 5 which is secured to the upper end of the fixed piece 4b of the radial bearing, thus forming a so-called thrust gap motor in which an axial gap is provided between the rotor magnet member 8 and the stator coil 5. It should be noted that, in this embodiment also, the movable pieces 3a and 4a of the thrust and radial bearings 3 and 4 may be joined together in the form of an integral bearing member in the same way as in the spindle motor shown in Fig. 6.

In any of the spindle motors shown in Figs. 5 to 7, the thrust bearing 3 is formed inwardly of the outer periphery of the rotor 6 and it is therefore possible to minimize the size of the base 1, obtain a compact spindle motor and hence reduce the installation area. In the case of a large-sized spindle motor of 5 to 8 inches level in diameter, the diameter and area of the thrust bearing can be increased even if it is disposed inwardly of the outer periphery of the rotor 6, and it is possible to obtain stable rotation of the rotor 6 by preloading the thrust bearing 3 by magnetic force in a counter direction to the dynamic pressure generated from the thrust bearing 3. In these embodiments also, the spindle motors are not necessarily limited to synchronous motors wherein the rotor magnet member comprises magnets but may be arranged in the form of induction motors wherein the rotor magnet member comprises iron cores, as in the embodiments shown in Figs. 1 to 4.

Fig. 8 shows an eighth embodiment of the spindle motor according to the present invention. A fixed piece 4b of a radial bearing 4 is secured to the outer periphery of the upper part of the support shaft 2, and a movable piece 4a of the radial bearing 4 is secured to the inner periphery of a cap-shaped rotor 6. The rotor 6 has a horizontally extending collar portion 17 formed at the lower end thereof. A movable piece 3a of a thrust bearing 3 is secured to the lower surface of the collar portion 17, and a fixed piece 3a of the thrust bearing 3 is secured to a base 1 in opposing relation to the movable piece 3a.

A rotor magnet member 8 is secured to the

rotor 6 at a position which is inward of the movable piece 3b of the thrust bearing 3. A stator coil 5 is secured to the base 1 in opposing relation to the rotor magnet member 8. The rotor magnet member 8 and the stator coil 5 has an axial gap therebetween, thereby forming a thrust gap motor.

The rotor 6 is arranged such that hard disks can be mounted on the outer peripheral portion of the rotor 6, and the thrust and radial bearings 3 and 4 are hydrodynamic bearings, in the same way as shown in the embodiment in Fig. 1.

In this embodiment, the fixed piece 4b of the radial bearing and the support shaft 2 may be formed in an integral structure, and the movable piece 4a of the radial bearing 4 and the rotor 6 may also be formed in an integral structure. Further, the movable piece 3a of the thrust bearing 3 and the collar portion 17 of the rotor 6 may be formed in an integral structure, and the fixed piece 3b of the thrust bearing 3 and the base 1 may also be formed in an integral structure.

In this embodiment, the length of the radial bearing 4 may be made substantially the same as the height of the rotor 6, thus, the radial bearing 4 can be increased. Such an arrangement enables enlargement of the effective dynamic pressure range within which sufficient load carrying capacity is provided, and hence permits minimization of the radial vibration. Since the radial bearing 4 is not formed in a cantilever structure, the starting torque is minimized. In addition, the radial bearing 4 is not required to be machined to any particularly high degree of accuracy. However, the length of the radial bearing 4 may be smaller than the height of the rotor 6.

When the spindle motor of this embodiment is used in a vertical position, the range of preload is the same as that expressed by the relationship (1) described above. When the spindle motor is used in a horizontal position, stable rotation is obtained if the rotor magnet member 8 is utilized to generate a force acting counter to the dynamic pressure to thereby subject the thrust bearing 3 to a force counter to the dynamic pressure, that is, preload the thrust bearing 3.

In this embodiment, the thrust bearing 3 is disposed outside the drive part and hence has a relatively large diameter, so that stable rotation is obtained. By sucking in air from the outer peripheral side of the thrust bearing 3, it is possible to prevent dust from scattering outwardly from the rotor magnet member 8 and the stator coil 5.

In a circumstances where dust causes no problem, it is, of course, possible to form the dynamic pressure generating grooves so as to generate dynamic pressure at the outside of the thrust bearing.

In the embodiment shown in Fig. 8, the size of each of the radial and thrust bearings 4 and 3 is increased to reduce the surface pressure acting on the contact surfaces, and the stator coil 5 is properly disposed to attain a structure which is free from local contact. Accordingly, if the members that constitute the bearings are made of, for example, a stainless steel, and a thin coat of lubricant is applied to the contact surfaces, it is possible to maintain a stable performance for a long period of time. When it is not possible to use any lubricant or the thickness of lubricant should be minimized if any, it is preferable to employ a ceramic material which is superior wear-resistant and low friction properties, particularly silicon carbide or alumina. In this spindle motor, a resilient pad 12 is interposed between the fixed piece 3b of the thrust bearing 3 and the base 1 to absorb any error in the perpendicularity between the thrust and radial bearings 3 and 4, and the force to preload the thrust bearing 3 can be obtained from the driving part, since the driving part is a thrust gap type.

Fig. 9 shows a ninth embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 8 but different therefrom in the following point. In the spindle motor shown in Fig. 9, the rotor magnet member 8 is secured to the inner peripheral of the lower part of the rotor 6 and the stator coil 5 is secured to the lower part of the support shaft 2, thereby forming a radial gap type motor. In the embodiment shown in Fig. 9, the center of the axial length of the stator coil 5 and that of the rotor magnet member 8 are offset from each other by a distance d, thereby enabling the thrust bearing 3 to be preloaded. The magnitude of the preload can be set at a desired value by varying the distance d.

Fig. 10 shows the structure of a tenth embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 9 but different therefrom in the following point. In this embodiment, the base 1 and the fixed piece 3b of the thrust bearing 3 are formed in an integral structure from the same constituent material, and the movable piece 3a of the thrust bearing 3, the rotor 6 and the movable piece 4a of the radial bearing 4 are formed in an integral structure from the same constituent material. The support shaft 2 is extended through a through-hole 16 which is provided in the upper end portion of the rotor 6, and the distal end of the support shaft 2 is secured to a stationary part (not shown). Since the support shaft 2 is supported at both ends by the base 1 and the stationary part, when the spindle motor is used in a horizontal position, it is possible to prevent deflection of the support shaft 2 which would otherwise be caused by the weight of the disks attached to the rotor 6. The arrangement what the support shaft 2 is supported at both ends

is advantageous for use in a large-sized spindle motor.

Fig. 11 shows the structure of an eleventh embodiment of the spindle motor according to the present invention. In this embodiment, a thrust bearing 3 which has a relatively small diameter is disposed between the upper end of a support shaft 2 and a cap-shaped rotor 6. A movable piece 3a of the thrust bearing 3 is secured to the inner surface of the upper part of the rotor 6 through a resilient pad 11. A movable piece 4a of a radial bearing 4 is secured to the inner peripheral surface of the rotor 6 below the movable piece 3a of the thrust bearing 3. A fixed piece 4b of the radial bearing 4 is concentrically secured to the outer peripheral surface of the upper part of the support shaft 2. The outer peripheral surface of the fixed piece 4b of the radial bearing 4 faces the inner peripheral surface of the movable piece 4a of the radial bearing 4. The upper end face of the fixed piece 4b defines a fixed piece 3b of the thrust bearing 3. The opposing cylindrical sliding surfaces of the movable and fixed pieces 4a and 4b of the radial bearing 4 are made of a ceramic material, and either of the cylindrical sliding surfaces has herringbone-shaped grooves C_1 for generating dynamic pressure, such as those shown in Fig. 13. Either of the opposing annular sliding surfaces of the movable and fixed pieces 3a and 3b of the thrust bearing 3 has spiral grooves C_2 for generating dynamic pressure, such as those shown in Fig. 14.

In the embodiment shown in Fig. 11, a stator coil 5 is secured to the support shaft 2 below the fixed piece 4b of the radial bearing 4. A rotor magnet member 8 is secured to the inner peripheral surface of the lower part of the rotor 6. The stator coil 5 and the rotor magnet member 8 has a radial gap therebetween to constitute a radial gap motor. By offsetting the axial center of the rotor magnet member 8 from that of the stator coil 5 by a distance d, the thrust bearing 3 is preloaded.

In the spindle motor structure of the embodiment shown in Fig. 11, the length of the radial bearing 4 can be increased and the dynamic pressure can therefore be increased, thereby enabling the spindle motor to operate with sufficient load carrying capacity, and thus permitting minimization of radial vibration. In addition, since the radial bearing 4 is not formed in a cantilever structure, the starting torque is minimized, and since a relatively high dynamic pressure is generated, the radial bearing 4 is not required to be machined to any particularly high degree of accuracy. When used in a horizontal position, the spindle motor is capable of stable rotation by virtue of the preload applied to the thrust bearing 3.

Fig. 12 shows the structure of a twelfth embodiment of the spindle motor according to the present invention, which is similar to the embodiment shown in Fig. 11 but different therefrom in the following point. In this embodiment, the stator coil 5 is secured to the upper surface of the base 1. The rotor magnet member 8 is secured to a bracket 13 which is secured to the inner peripheral surface of the lower part of the rotor 6. The stator coil 5 and the rotor magnet member 8 have an axial gap therebetween to constitute a thrust gap motor. In the embodiment shown in Fig. 12, the magnetic force acting between the stator coil 5 and the rotor magnet member 8 can be utilized to preload the thrust bearing 3 in the counter direction to the dynamic pressure generated therefrom.

The spindle motor of the present invention provides the following advantageous effects:

(1) Since the constituent members of the radial bearing are formed to be integral with the associated constituent members of the spindle motor, alignment effected at the time of assembling is facilitated. In addition, it is easy to carry out precise machining of the radial bearing itself.

(2) Since the radial bearing is made relatively long to bear the rotor over a predetermined range which includes at least the center of gravity of the rotor, the spindle motor is capable of operating with sufficient load carrying capacity. Accordingly, the dynamic pressure increases and the radial vibration is minimized. Since the constituent members of the radial bearing are not required to be machined to any particularly high degree of accuracy, the product cost lowers.

(3) By disposing the thrust bearing at the outer periphery of the rotor, it is possible to increase both the diameter and area of the thrust bearing and hence obtain a high dynamic pressure. In addition, by magnetically preloading the thrust bearing in the thrust direction, the inclination of the support shaft with respect to the radial bearing is corrected and the rotor is capable of stably rotating without being sprung out by the dynamic pressure applied thereto in the thrust direction. In particular, even when the spindle motor is used in a horizontal position, the rotor rotates stably by virtue of the cooperation of the elongated radial bearing and the preload applied to the thrust bearing.

(4) By forming the thrust bearing at a position which is inward of the outer periphery of the rotor, the spindle motor can be made simple and compact, so that the area required for installation decreases and the field of its application expands.

(5) By forming the thrust and radial bearings from a ceramic material, an oil free type or a minimal lubricant type spindle motor is obtained.

(6) By interposing a resilient pad between the thrust bearing and the base, machining errors of the base and the bearing can be compensated and it is therefore possible to maintain an excellent

sliding condition.

(7) By forming a thrust gap type spindle motor, it is possible to eliminate a moment which would otherwise be generated due to the imbalance of radial magnetic force acting between the stator coil and the rotor magnet member and hence enable the motor to rotate with minimal vibrations.

(8) In the case of a large-sized spindle motor having a diameter of 5 to 8 inches, for example, even if the thrust bearing is disposed inwardly of the outer periphery of the rotor, the diameter and area of the thrust bearing can be increased, and by magnetically preloading the thrust bearing in the thrust direction, the inclination of the radial bearing is corrected and the rotor is capable of stably rotating without being sprung out by the dynamic pressure applied thereto in the thrust direction.

Claims

1. A spindle motor comprising a stator including a support shaft positioned on a base, a cap-shaped rotor rotatably and concentrically disposed around said support shaft, thrust and radial bearings disposed between said stator and said rotor, a stator coil secured to said stator, and a rotor magnet member secured to said rotor in opposing relation to said stator coil,
wherein said thrust and radial bearings are hydrodynamic bearings, said thrust bearing comprising movable and fixed pieces, said movable piece being secured to the lower end of a cylindrical portion of said rotor and extended outwardly from said cylindrical portion, said fixed piece being secured to said base in opposing relation to said movable piece, said stator coil being secured to the outer peripheral portion of said support shaft above said radial bearing, and said rotor magnet member being secured to the inner peripheral surface of said rotor so that a radial gap is provided between said stator coil and said rotor magnet member.

2. A spindle motor comprising a stator including a support shaft stood on a base, a cap-shaped rotor rotatably and concentrically disposed around said support shaft, thrust and radial bearings disposed between said stator and said rotor, a stator coil secured to said stator, and a rotor magnet member secured to said rotor in opposing relation to said stator coil,
wherein said thrust and radial bearings are hydrodynamic bearings, said thrust bearing comprising movable and fixed pieces, said movable piece being secured to the lower end of a cylindrical portion of said rotor and extended outwardly from said cylindrical portion, said fixed piece being secured to said base in opposing relation to said movable piece, said stator coil being secured to the outer peripheral portion of said support shaft above said radial bearing, and said rotor magnet member being secured to the ceiling of said rotor so that an axial gap is provided between said stator coil and said rotor magnet member.

3. A spindle motor comprising a stator including a support shaft stood on a base, a cap-shaped rotor rotatably and concentrically disposed around said support shaft, thrust and radial bearings disposed between said stator and said rotor, a stator coil secured to said stator, and a rotor magnet member secured to said rotor in opposing relation to said stator coil,
wherein said thrust and radial bearings are hydrodynamic bearings, said thrust bearing comprising movable and fixed pieces, said movable piece being secured to the lower end of a cylindrical portion of said rotor and extended inwardly from said cylindrical portion, said fixed piece being secured to said base in opposing relation to said movable piece, and both said stator coil and said rotor magnet member being disposed above said radial bearing.

4. A spindle motor according to Claim 3, wherein said rotor magnet member is secured to the ceiling of said rotor so that an axial gap is provided between said stator coil and said rotor magnet member.

5. A spindle motor comprising a stator including a support shaft positioned on a base, a cap-shaped rotor rotatably and concentrically disposed around said support shaft, thrust and radial bearings disposed between said stator and said rotor, a stator coil secured to said stator, and a rotor magnet member secured to said rotor in opposing relation to said stator coil, wherein said thrust and radial bearings are hydrodynamic bearings, said thrust bearing comprising movable and fixed pieces, said movable piece being secured to the lower end of a cylindrical portion of said rotor and extended outwardly from said cylindrical portion, said fixed piece being secured to said base in opposing relation to said movable piece, and said rotor magnet member being secured to said rotor at a position which is inward of said movable piece of said thrust bearing.

6. A spindle motor according to Claim 5, wherein said stator is secured to said base so that an axial gap is provided between said stator coil and said rotor magnet member.

7. A spindle motor according to Claim 5, wherein said stator coil is secured to the lower part of said support shaft so that a radial gap is provided between said stator coil and said rotor magnet member.

8. A spindle motor comprising a stator including a support shaft positioned on a base, a cap-

shaped rotor rotatably and concentrically disposed around said support shaft, thrust and radial bearings disposed between said stator and said rotor, a stator coil secured to said stator, and a rotor magnet member secured to said rotor in opposing relation to said stator coil,
wherein said thrust and radial bearings are hydrodynamic bearings, said thrust bearing being disposed above said radial bearing, and both said stator coil and said rotor magnet member being disposed below said radial bearing.

9. A spindle motor according to Claim 8, wherein said stator coil is secured to said base so that an axial gap is provided between said stator coil and said rotor magnet member.

10. A spindle motor according to any one of Claims 1 to 9, wherein said thrust bearing is preloaded by a magnetic force acting counter to a dynamic pressure acting in the direction of thrust.

11. A spindle motor according to any one of Claims 1, 3, 5, 7 and 8, wherein the axial center of said rotor magnet member is offset from the axial center of said stator coil by a predetermined amount in the counter direction to the dynamic pressure generated from said thrust bearing, thereby preloading said thrust bearing in a counter direction to the dynamic pressure acting in the direction of thrust.

12. A spindle motor according to any one of Claims 1 to 9, wherein said radial bearing is disposed so as to bear said rotor over a predetermined range including the center of gravity of said rotor.

13. A spindle motor according to any one of Claims 1 to 9, wherein said fixed piece of said thrust bearing is secured to said stator through a resilient pad.

14. A spindle motor according to Claim 13, wherein said resilient pad is silicone rubber.

15. A spindle motor according to any one of Claims 1 to 9, wherein said movable piece of said thrust bearing is secured to said rotor through a resilient pad.

16. A spindle motor according to Claim 15, wherein said resilient pad is silicone rubber.

17. A spindle motor according to any one of Claims 1 to 9, wherein the opposing annular sliding surfaces of said fixed and movable pieces of said thrust bearing are made of a ceramic material or made of a member coated with a kind of material different from the groundwork thereof or a member having a treated surface of degenerated groundwork thereof, and either of said sliding surfaces has spiral grooves for generating dynamic pressure.

18. A spindle motor according to Claim 17, wherein said ceramic material is either silicon carbide or alumina.

19. A spindle motor according to any one of Claims 1 to 9, wherein the opposing cylindrical sliding surfaces of said fixed and movable pieces of said radial bearing are made of a ceramic material material or made of a member coated with a kind of material different from the groundwork thereof or a member having a treated surface of degenerated groundwork thereof, and either of said cylindrical sliding surfaces has herringbone-shaped grooves for generating dynamic pressure.

20. A spindle motor according to Claim 19, wherein said ceramic material is either silicon carbide or alumina.

21. A spindle motor according to any one of Claims 1 to 4, 8 and 9, wherein some or all of said movable piece of said radial bearing, said rotor and said movable piece of said thrust bearing are arranged in an integral structure.

22. A spindle motor according to Claim 21, either one or both of the moving pieces and the radial and thrust bearings in the integral structure are coated with a kind of material different from that of groundwork thereof, or the groundwork thereof is surface treated.

23. A spindle motor according to any one of Claims 1, 2, 8 and 9, wherein some or all of said fixed piece of said radial bearing, said support shaft, said fixed piece of said thrust bearing and said base are arranged in an integral structure.

24. A spindle motor according to Claim 23, either one or both of the fixed pieces of the radial and thrust bearings in the integral structure are coated with a kind of material different from that of groundwork thereof, or groundwork thereof is surface treated.

25. A spindle motor according to any one of Claims 1 to 9, wherein said rotor has a support member adapted to hold hard disks on the outer peripheral surface thereof.

26. A spindle motor according to any one of Claims 1 to 9, wherein said support shaft is extended through a through-hole provided in the upper end portion of said rotor, and said support shaft being loosely fitted in said through-hole.

27. A spindle motor according to Claim 26, wherein the distal end of said support shaft is secured to a stationary part.

28. A spindle motor according to any one of Claims 1 to 9, wherein the upper end of said rotor is closed and not pierced by said support shaft.

Fig. 1

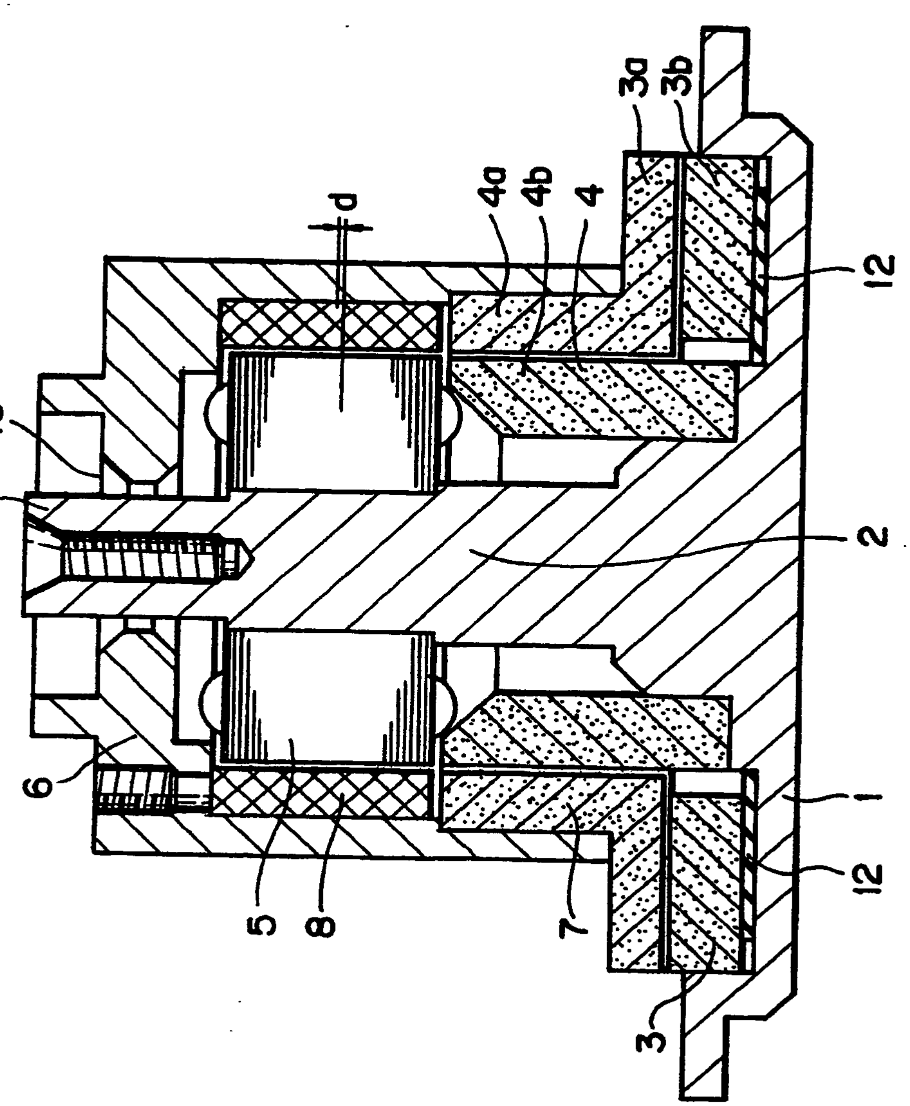


Fig. 2

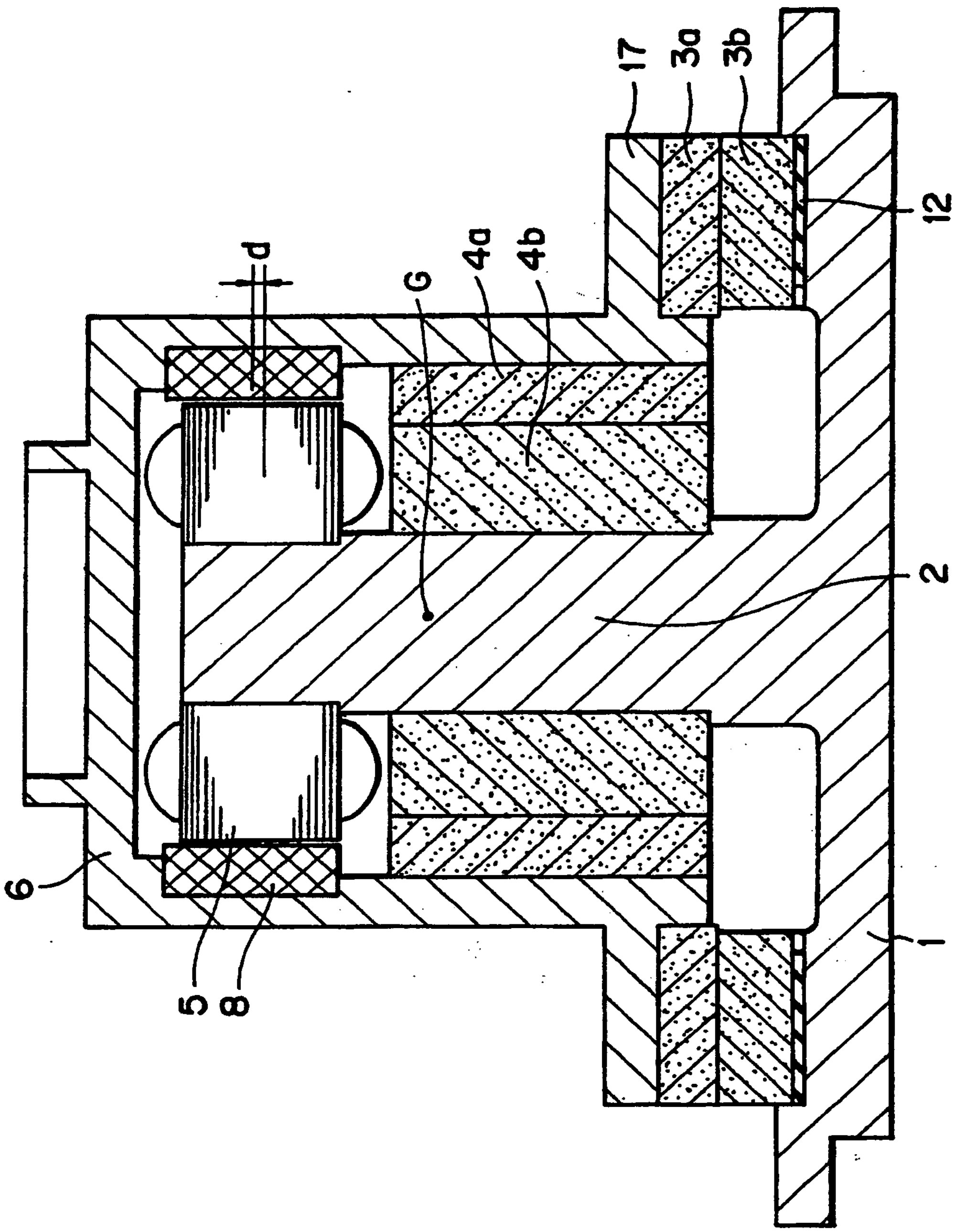


Fig. 4

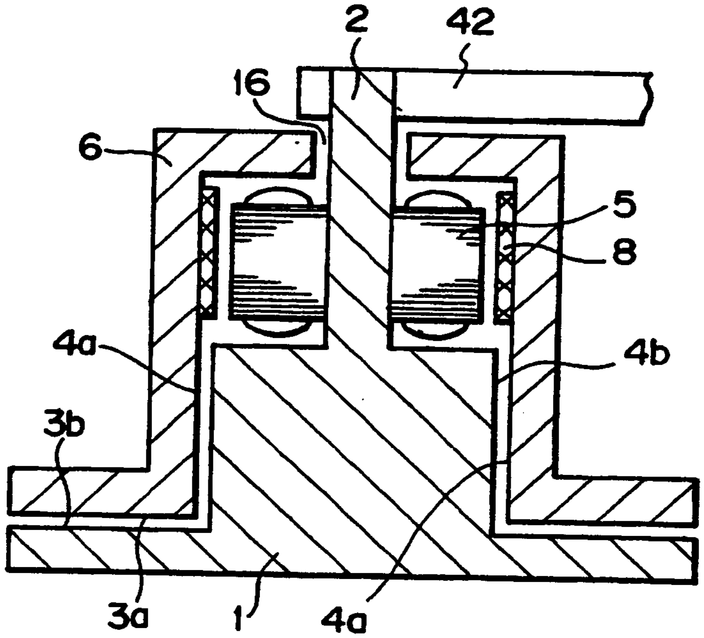


Fig. 5

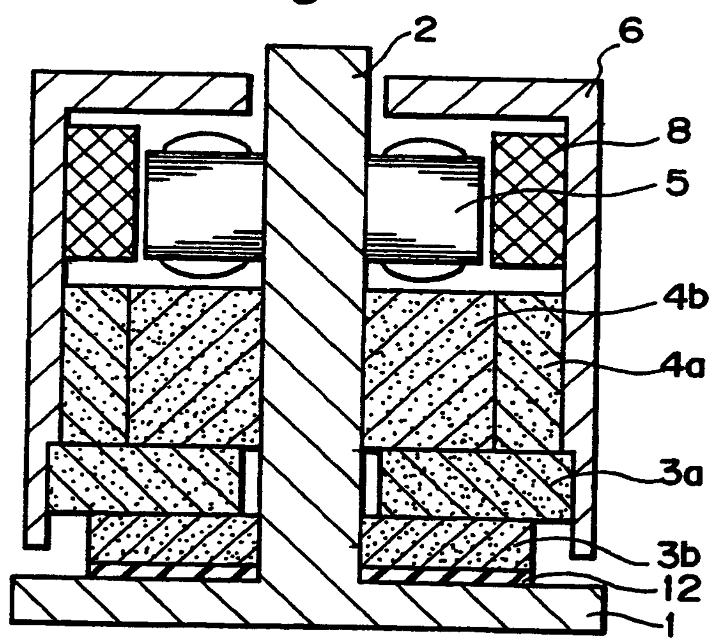


Fig. 4

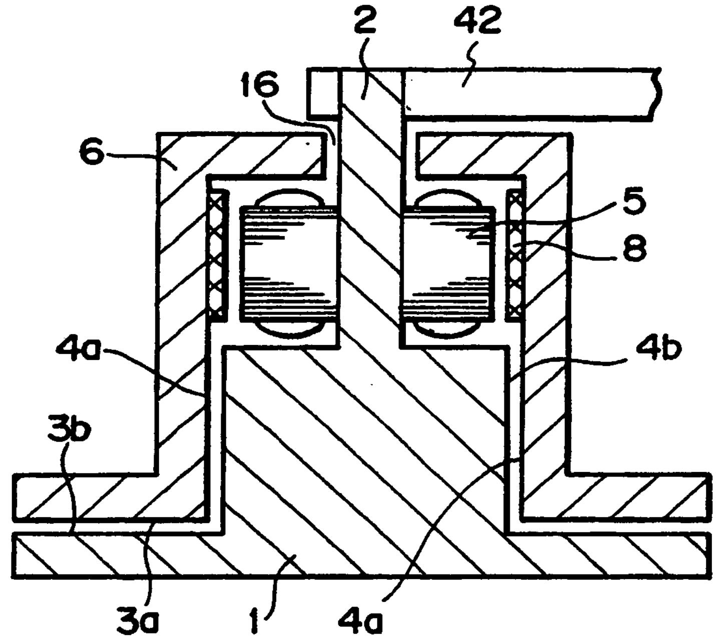


Fig. 5

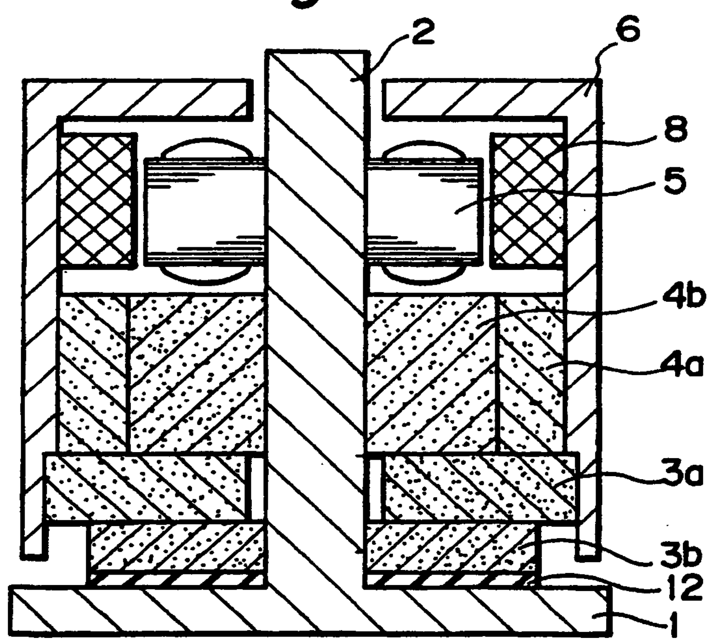


Fig. 6

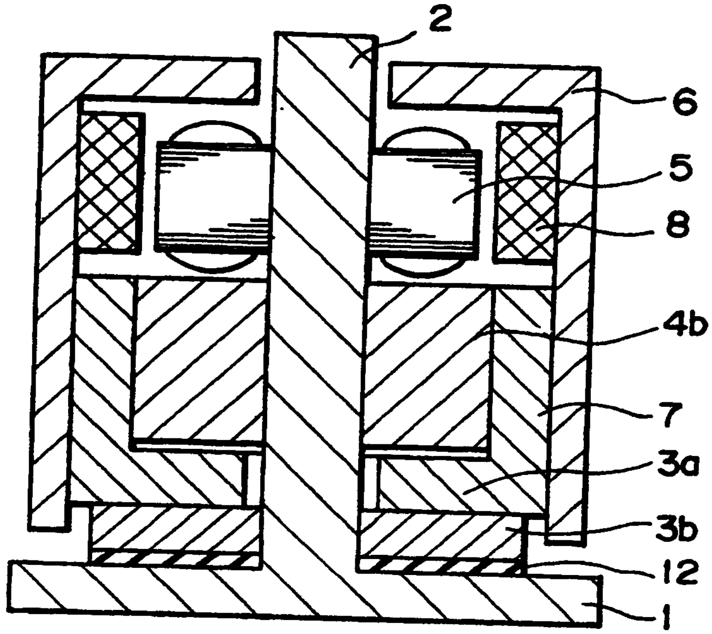


Fig. 7

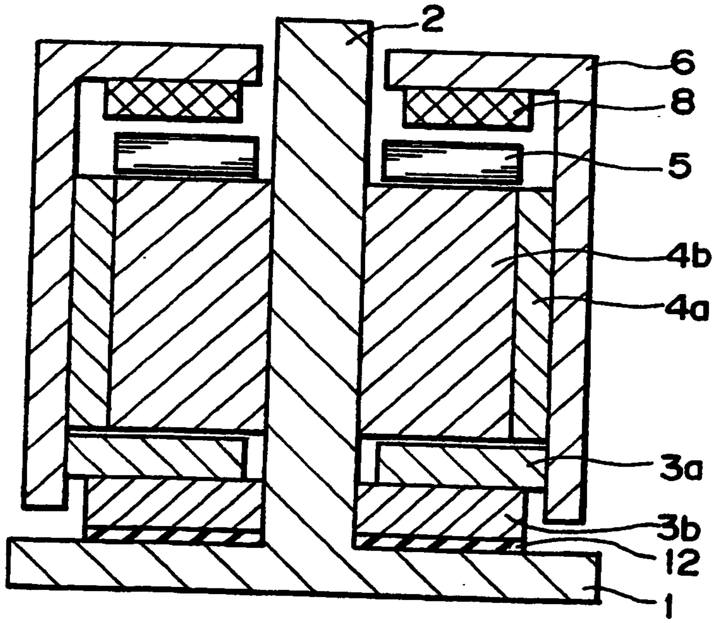


Fig. 8

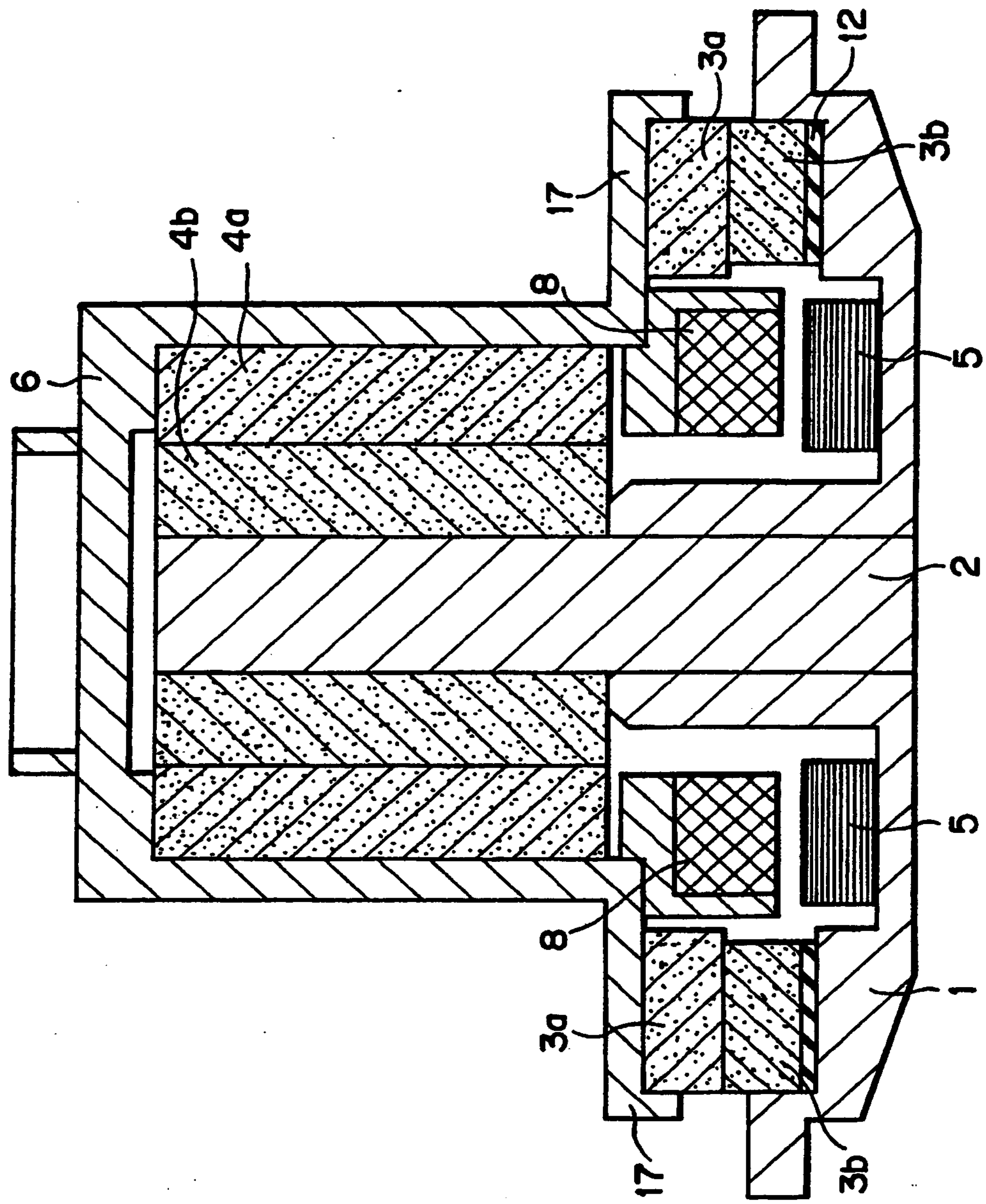


Fig. 9

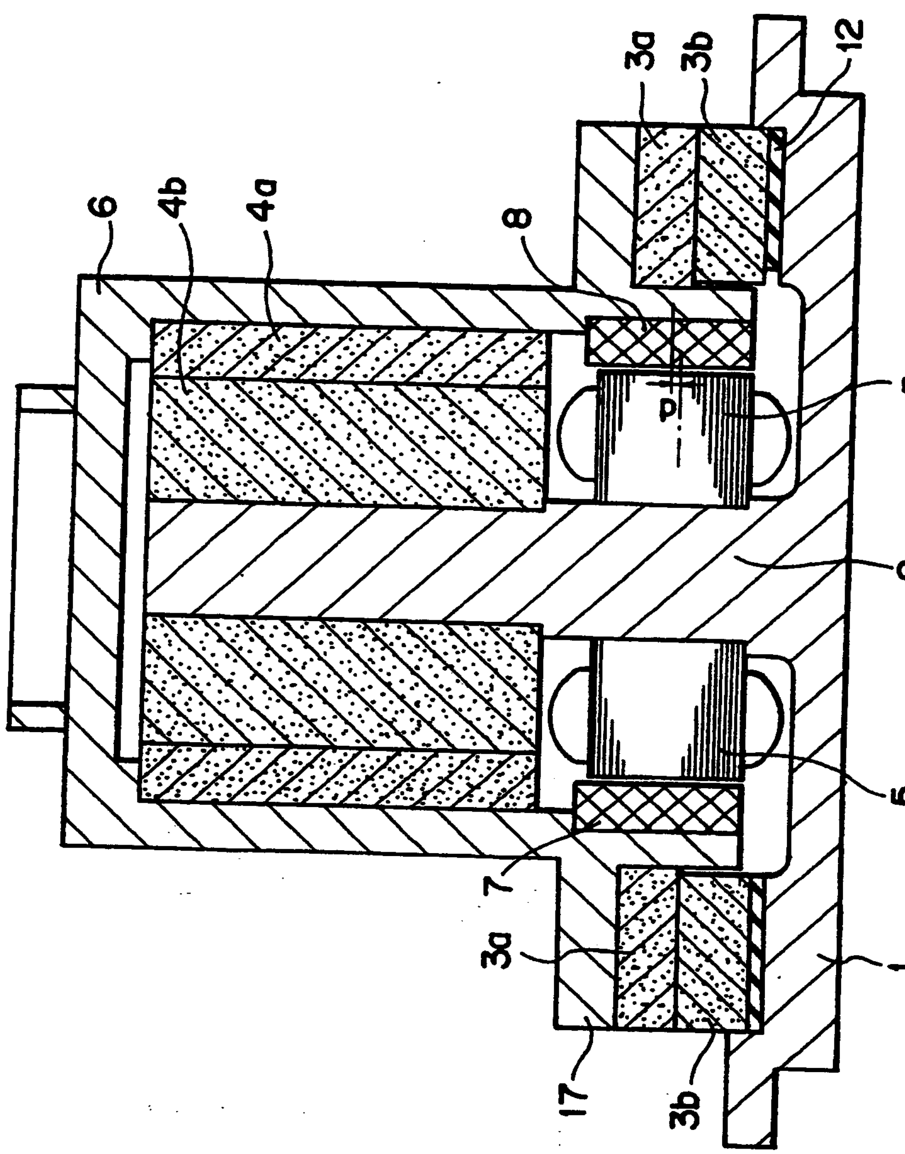


Fig. 10

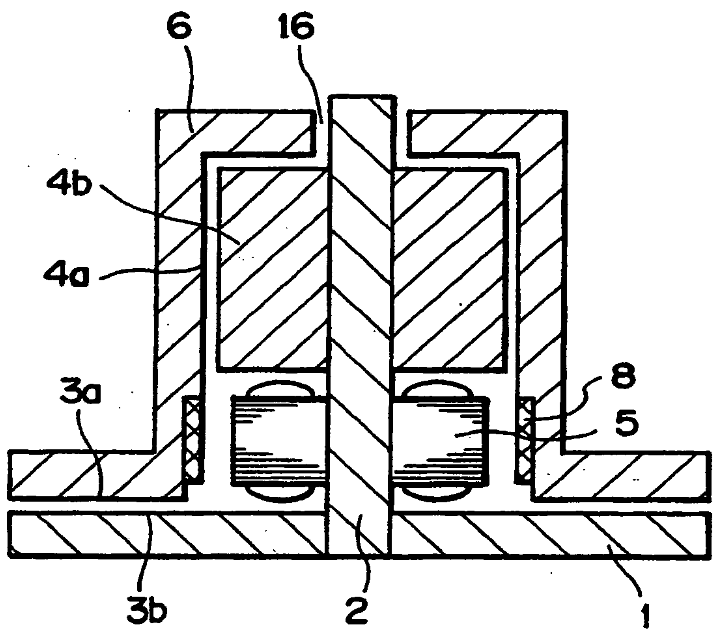


Fig. 11

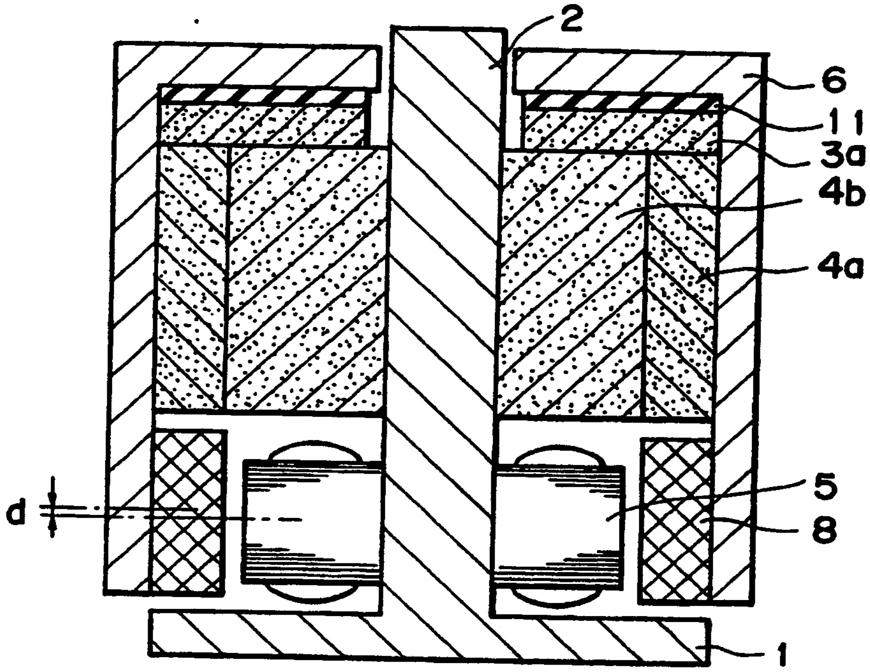


Fig. 12

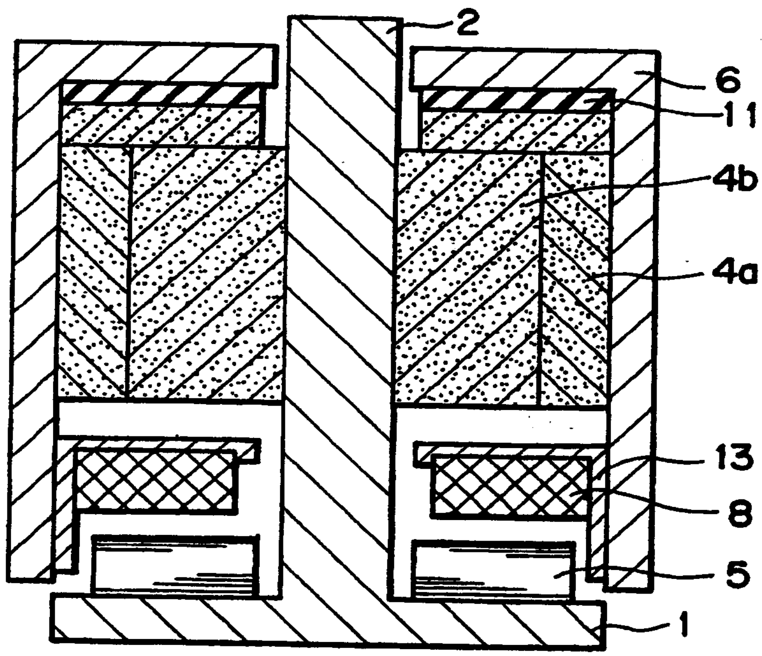


Fig. 13

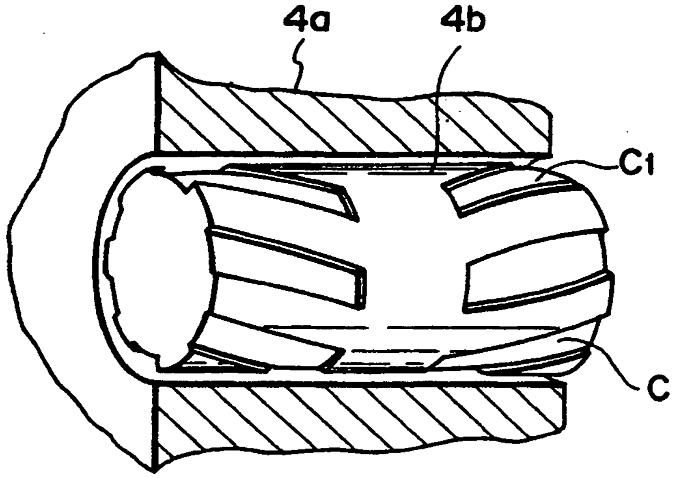


Fig. 14

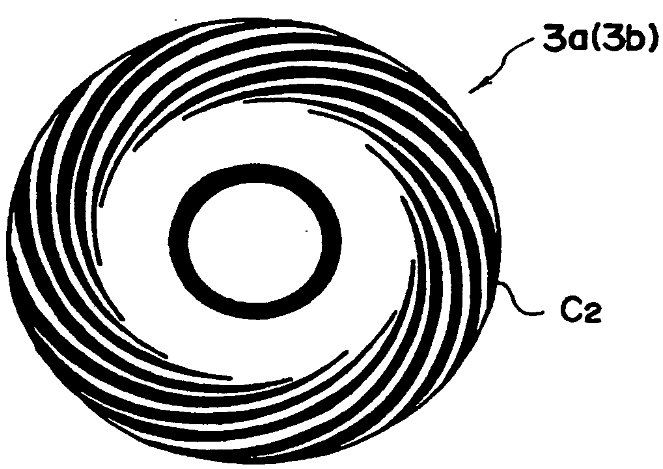


Fig. 15

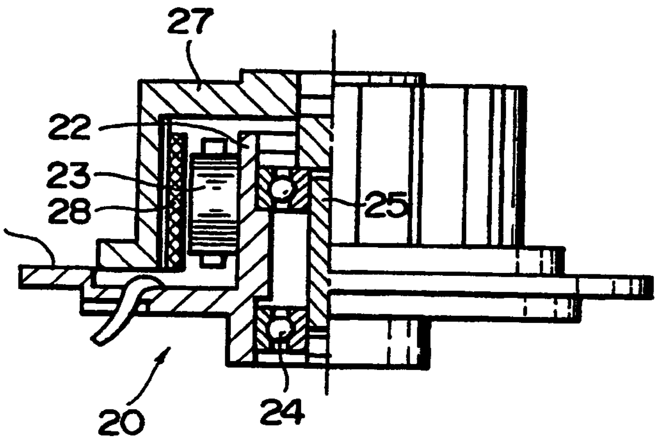
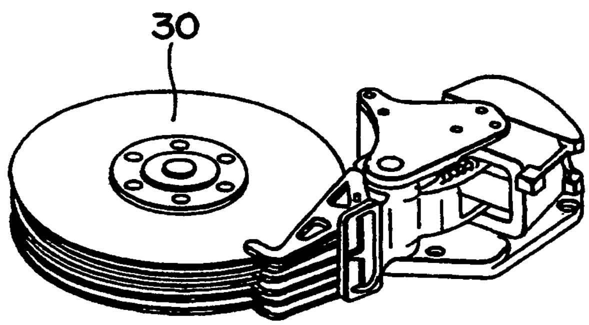


Fig. 16



(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: **0 392 500 A3**

(12) EUROPEAN PATENT APPLICATION

(21) Application number: 90106941.9

(22) Date of filing: 11.04.90

(51) Int. Cl.5: G11B 19/20, H02K 7/04

(30) Priority: 12.04.89 JP 92161/89
12.07.89 JP 179647/89
08.08.89 JP 205077/89
30.08.89 JP 223679/89

(43) Date of publication of application:
17.10.90 Bulletin 90/42

(84) Designated Contracting States:
AT BE CH DE ES FR GB IT LI NL SE

(88) Date of deferred publication of the search report:
02.05.91 Bulletin 91/18

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(54) Spindle motor.

(57) A spindle motor comprising a stator (5) including a support shaft (2) positioned on a base, a cap-shaped rotor (6) rotatably and concentrically disposed around the support shaft, thrust and radial bearings (3, 4) disposed between the stator and the rotor, a stator coil (5) secured to the stator, and a rotor magnet member (6) secured to the rotor in opposing relation to the stator coil. The thrust and radial bearings are hydrodynamic bearings. A movable piece that constitutes a part of the thrust bearing is secured to the lower end of a cylindrical portion of the rotor and extended outwardly or inwardly from the cylindrical portion, or is disposed above the rotor. The stator coil (5) is secured to the outer peripheral portion of the support shaft above or below the radial bearing. The rotor magnet member (6) is secured to the inner peripheral surface or ceiling of the rotor. Either a radial or axial gap is provided between the stator coil and the rotor magnet member.

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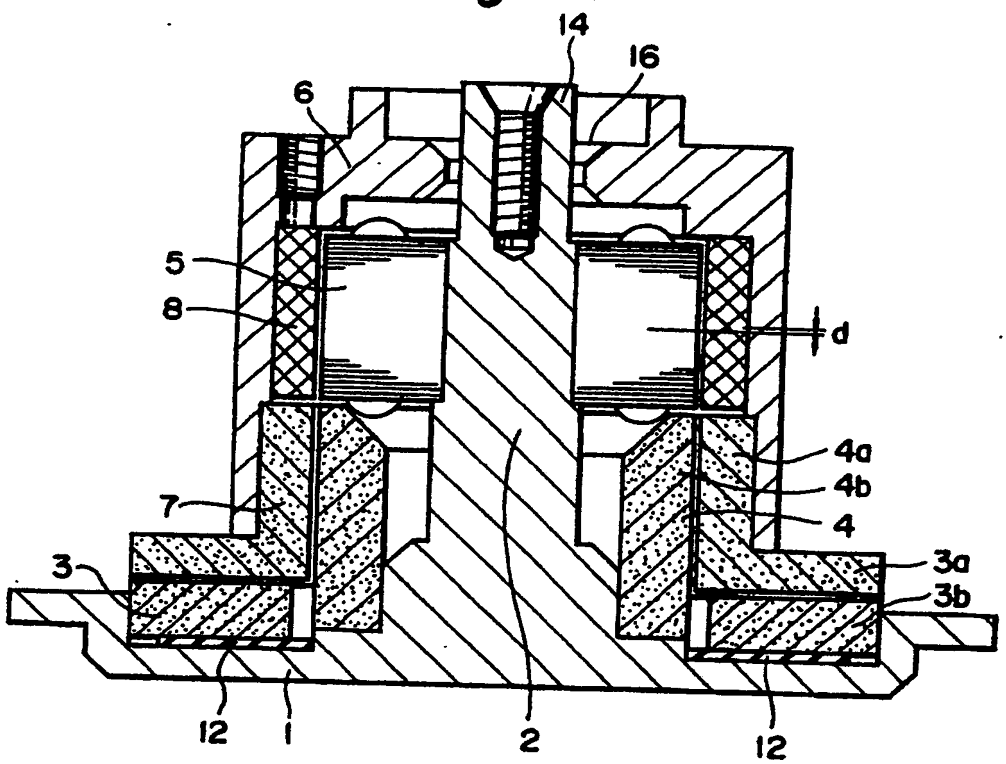
Fig. 1



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EUROPEAN SEARCH REPORT

Application Number

EP 90 10 6941

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
Y	GB-A-1524662 (THE UNIVERSITY OF SOUTHAMPTON) * the whole document * ---	1-6, 8-11, 17-25	G11B19/20 H02K7/04
Y	PATENT ABSTRACTS OF JAPAN vol. 10, no. 150 (E-408)(2207) 31 May 1986, & JP-A-61 009138 (CANON K.K.) 16 January 1986, * the whole document * ---	1, 10, 11	
P,Y	PATENT ABSTRACTS OF JAPAN vol. 13, no. 170 (M-817) 21 April 1989, & JP-A-01 003315 (NIPPON SEIKO K.K.) 09 January 1989, * the whole document *	2	
Y	---	2	
Y	PATENT ABSTRACTS OF JAPAN vol. 8, no. 181 (E-261)(1618) 21 August 1984, & JP-A-59 072956 (TOSHIBA K.K.) 25 April 1984, * the whole document * ---	3, 4	
P,Y	FR-A-2627313 (SOCIETE DE MECANIQUE MAGNETIQUE S.A.) * abstract; figures 1, 2 * ---	5, 6	TECHNICAL FIELDS SEARCHED (Int. Cl.5) G11B H02K
Y	PATENT ABSTRACTS OF JAPAN vol. 8, no. 169 (P-292) 04 August 1984, & JP-A-59 063024 (MATSUSHITA DENKI SANGYO KABUSHIKI KAISHA) 10 April 1984, * the whole document * ---	8-10	
Y	PATENT ABSTRACTS OF JAPAN vol. 9, no. 245 (M-418) 02 October 1985, & JP-A-60 098213 (CANON K.K.) 01 June 1985, * the whole document * ---	17-25	
A	PATENT ABSTRACTS OF JAPAN vol. 9, no. 65 (E-304)(1788) 26 March 1985, & JP-A-59 204441 (CANON K.K.) 19 November 1984, * the whole document *	9, 10	

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	28 FEBRUARY 1991	DEVERGRANNE C.

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X : particularly relevant if taken alone
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Application Number

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Page 2

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Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
A	PATENT ABSTRACTS OF JAPAN vol. 7, no. 178 (E-191)(1323) 06 August 1983, & JP-A-58 083552 (HITACHI KOKI K.K.) 19 May 1983, * the whole document *		
A	US-A-4726640 (AKIHIKO IWAMA ET AL)		
			TECHNICAL FIELDS SEARCHED (Int. Cl.5)

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	28 FEBRUARY 1991	DEVERGRANNE C.

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L : document cited for other reasons

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EP30311 ②

US006371650B1

(12) **United States Patent**
Goto et al.

(10) **Patent No.: US 6,371,650 B1**
(45) **Date of Patent: Apr. 16, 2002**

(54) **HYDRAULIC DYNAMIC BEARING AND SPINDLE MOTOR AND ROTARY ASSEMBLY PROVIDED**

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/414,244**

(22) Filed: **Oct. 7, 1999**

(30) **Foreign Application Priority Data**

Oct. 8, 1998	(JP)	10-286881
Oct. 8, 1998	(JP)	10-286894
Oct. 15, 1998	(JP)	10-294272
Oct. 15, 1998	(JP)	10-294273
Oct. 1, 1999	(JP)	10-280978

(51) **Int. Cl.**[7] **F16C 32/06**
(52) **U.S. Cl.** **384/110**; 384/113; 384/115; 384/118
(58) **Field of Search** 384/107, 110, 384/113, 115, 118

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(57) **ABSTRACT**

A double sleeve type dynamic bearing comprises a fixed shaft having at least one end fixedly mountable to an apparatus in which the bearing is utilized, a rotary sleeve arranged coaxially with the fixed shaft so that a first fine gap is formed therebetween, a fixed sleeve arranged coaxially with the rotary sleeve so that a second fine gap is formed therebetween, and a lubrication oil filled in the fine gaps. The first fine gap and the second fine gap each have an open end exposed to air outside the bearing and an opposite end that is not exposed to the air, the opposite ends being in communication with each other. A holding member holds the fixed shaft and the fixed sleeve and is disposed adjacent to a lower end surface of the rotary sleeve to form a third fine gap between the holding member and the lower end surface of the rotary sleeve. The third fine gap is formed with a thrust dynamic pressure producing groove, and opposite ends of the first and second fine gaps meet each other through the third fine gap. A peripheral surface of at least one of the fixed shaft, the rotary sleeve and the fixed sleeve forming at least one of the first and second fine gaps has a dynamic pressure producing groove formed therein.

46 Claims, 14 Drawing Sheets

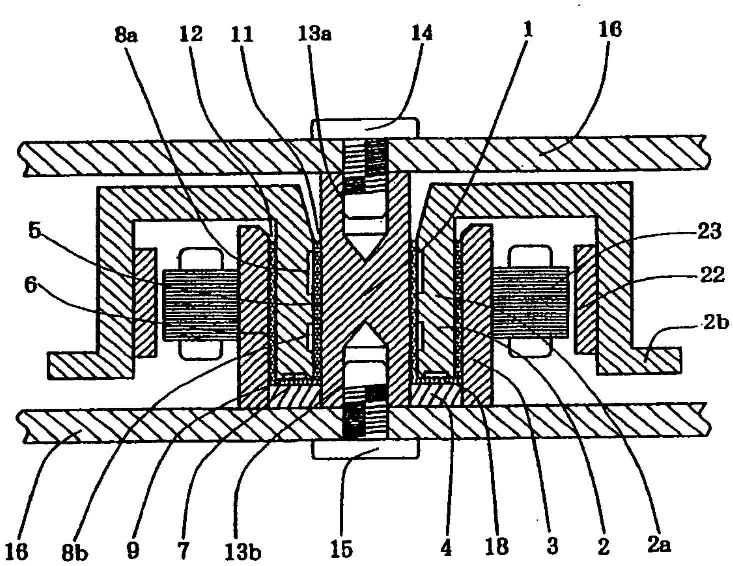


FIG. 1

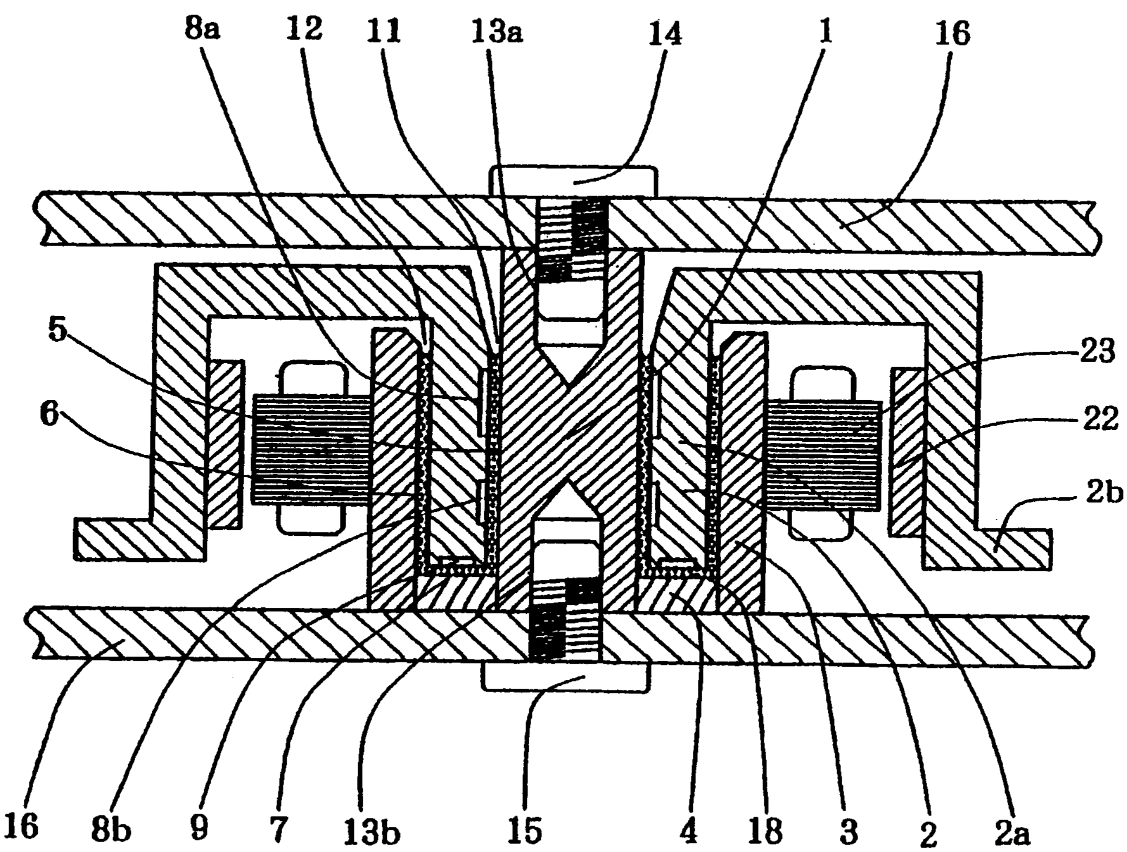


FIG. 2

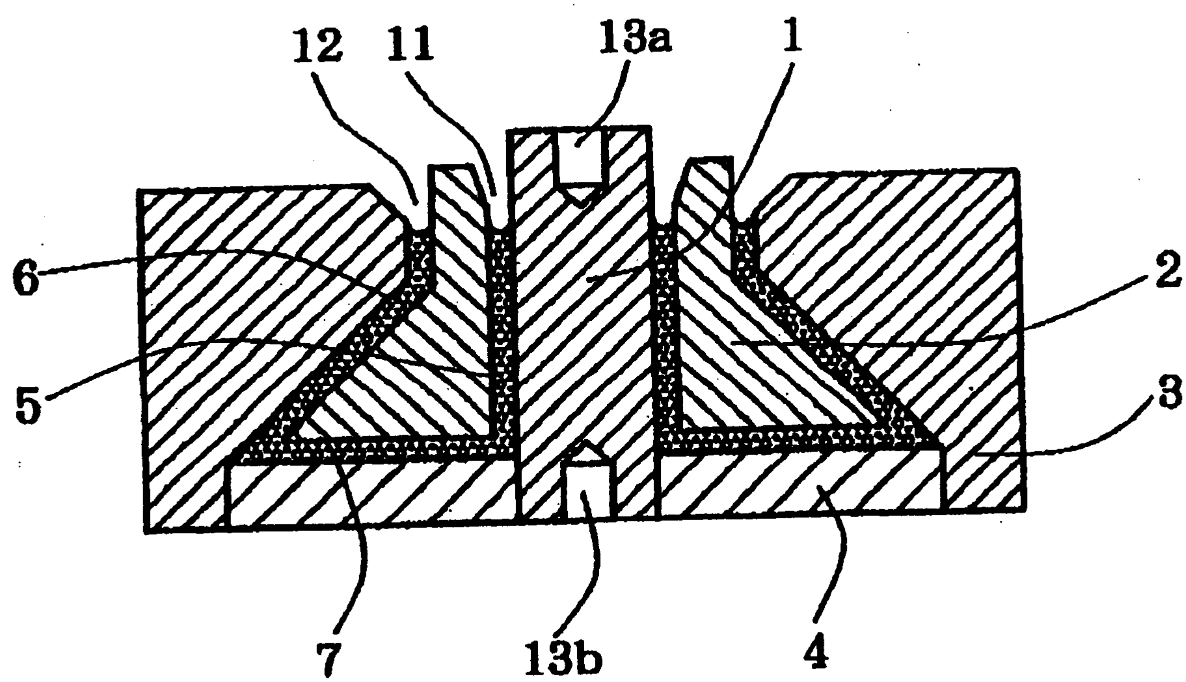


FIG. 3

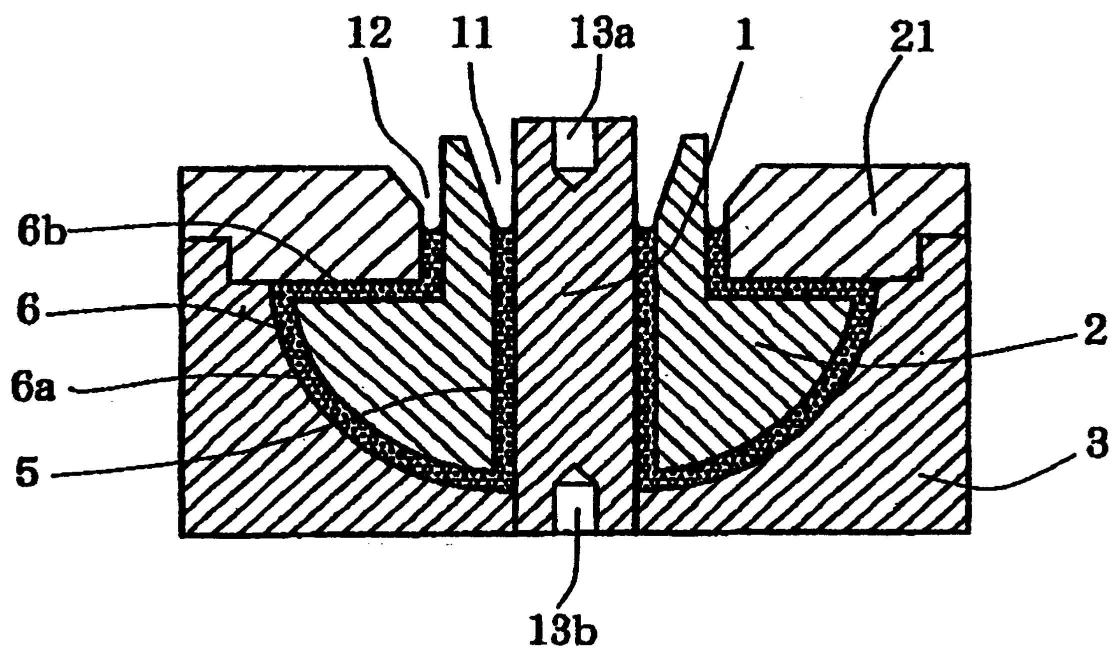


FIG. 4

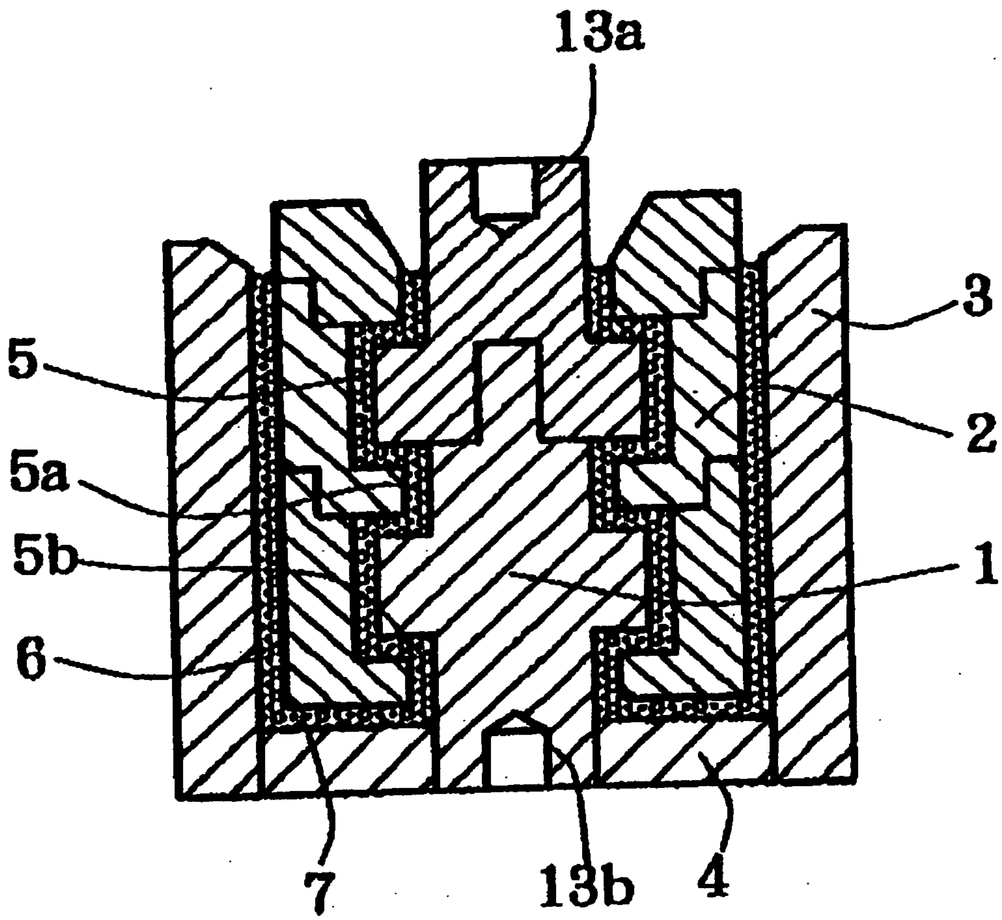


FIG. 5

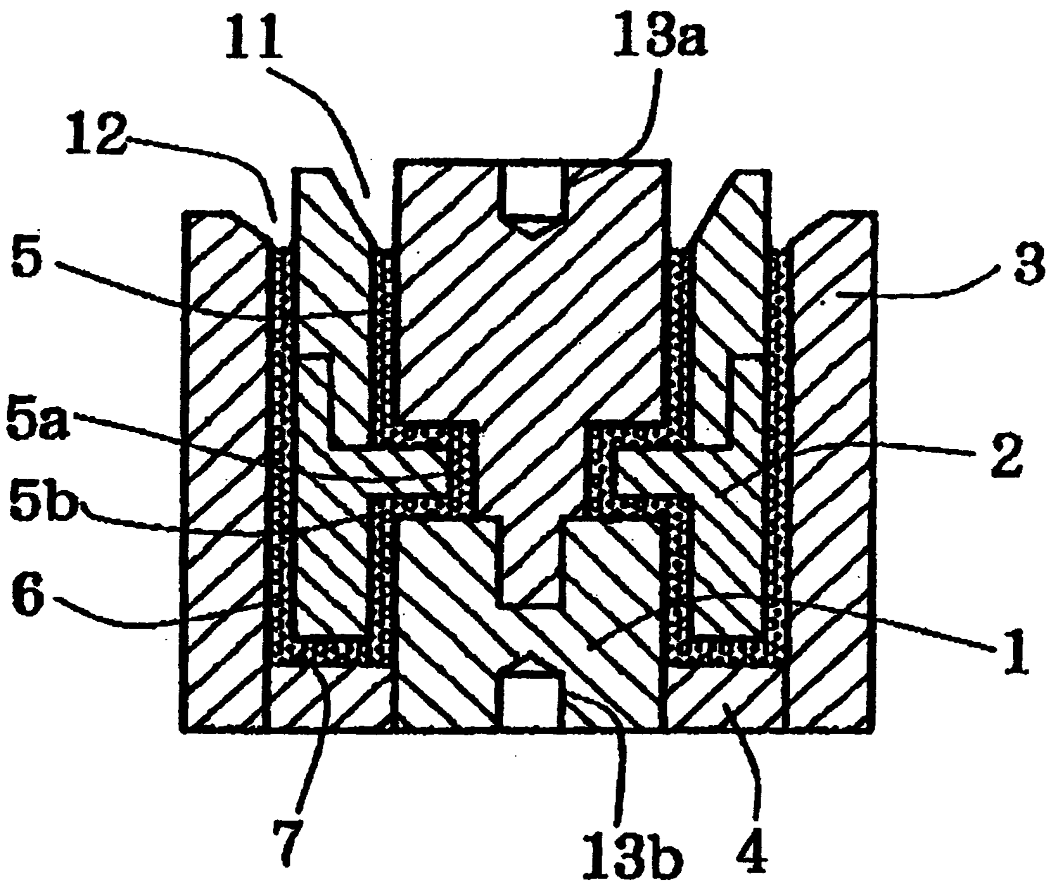


FIG. 6

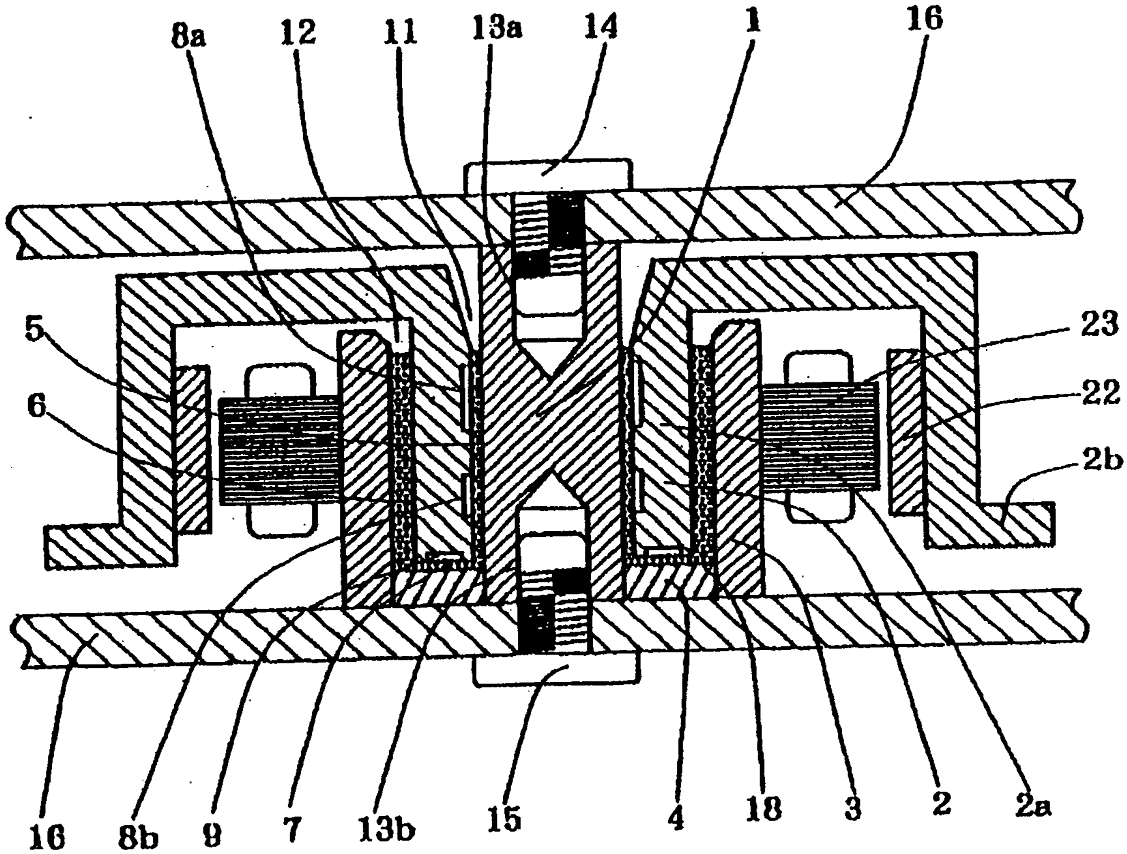


FIG. 7

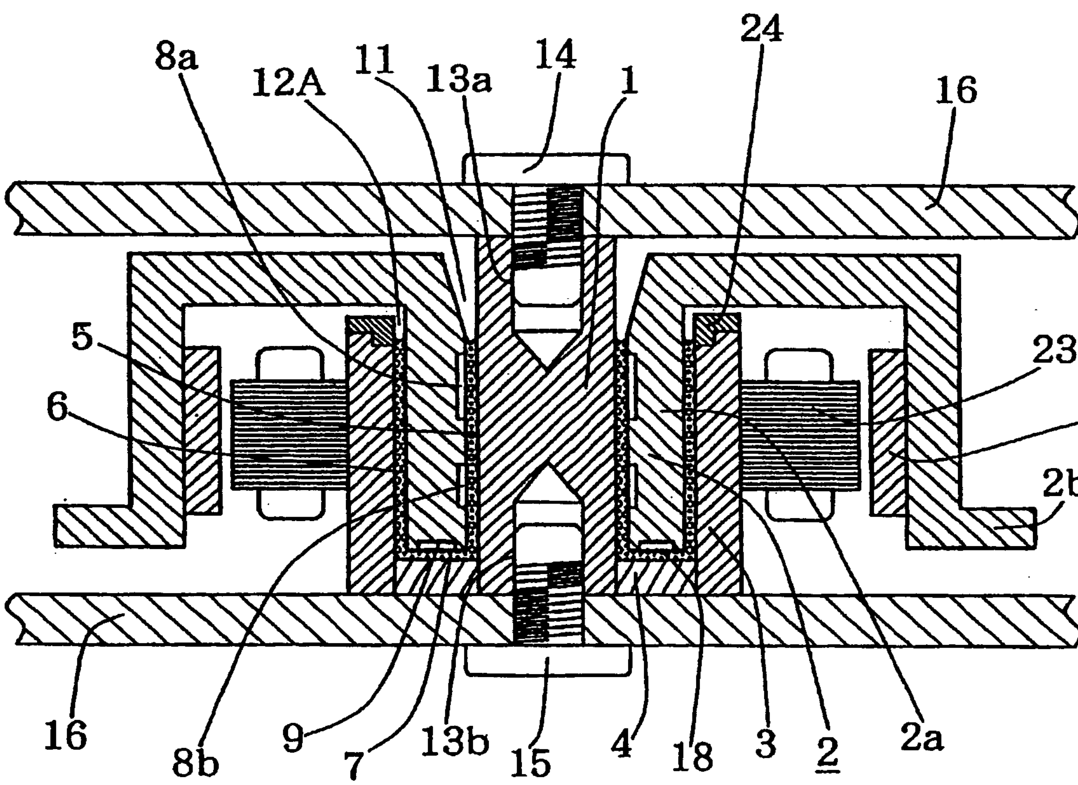


FIG. 8

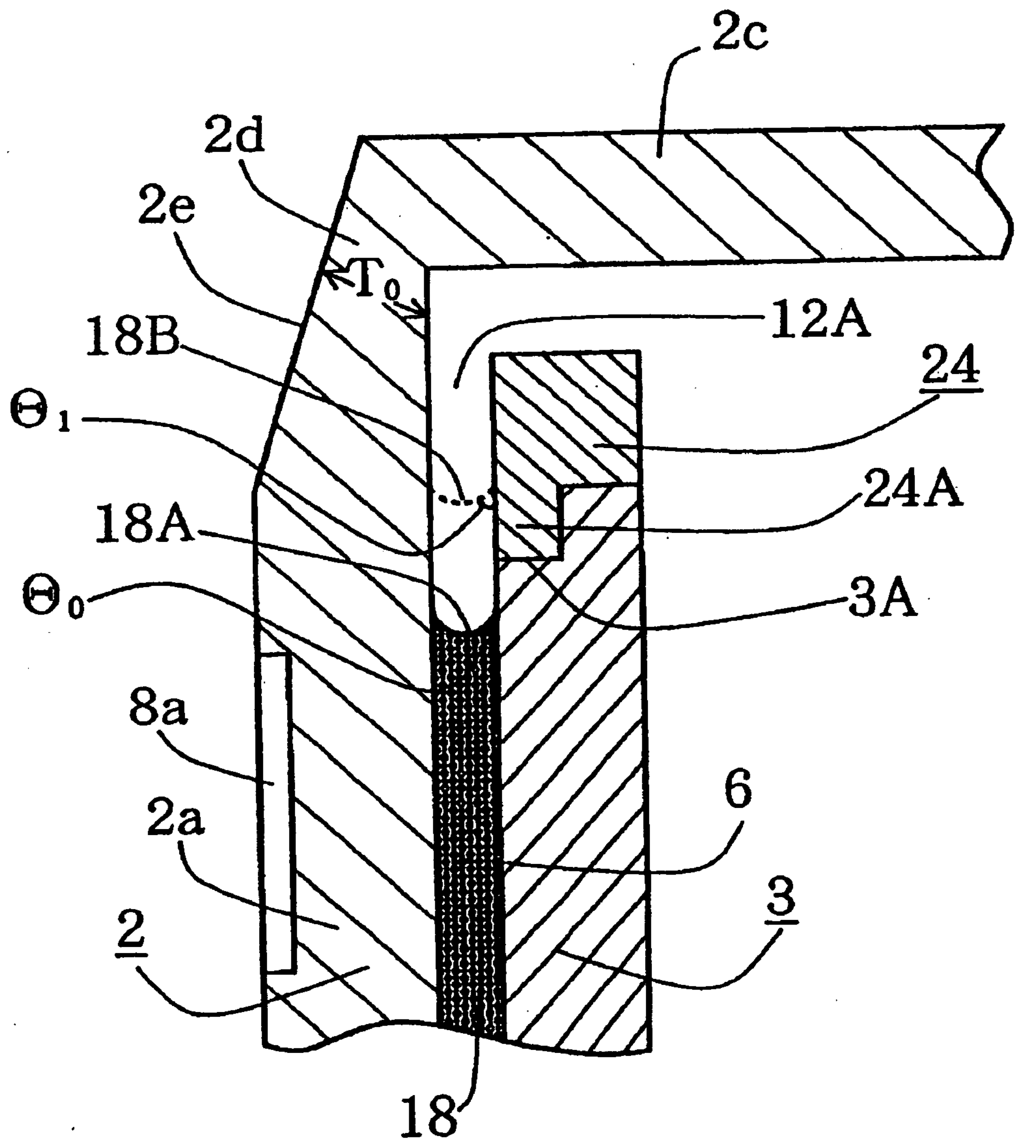


FIG. 9

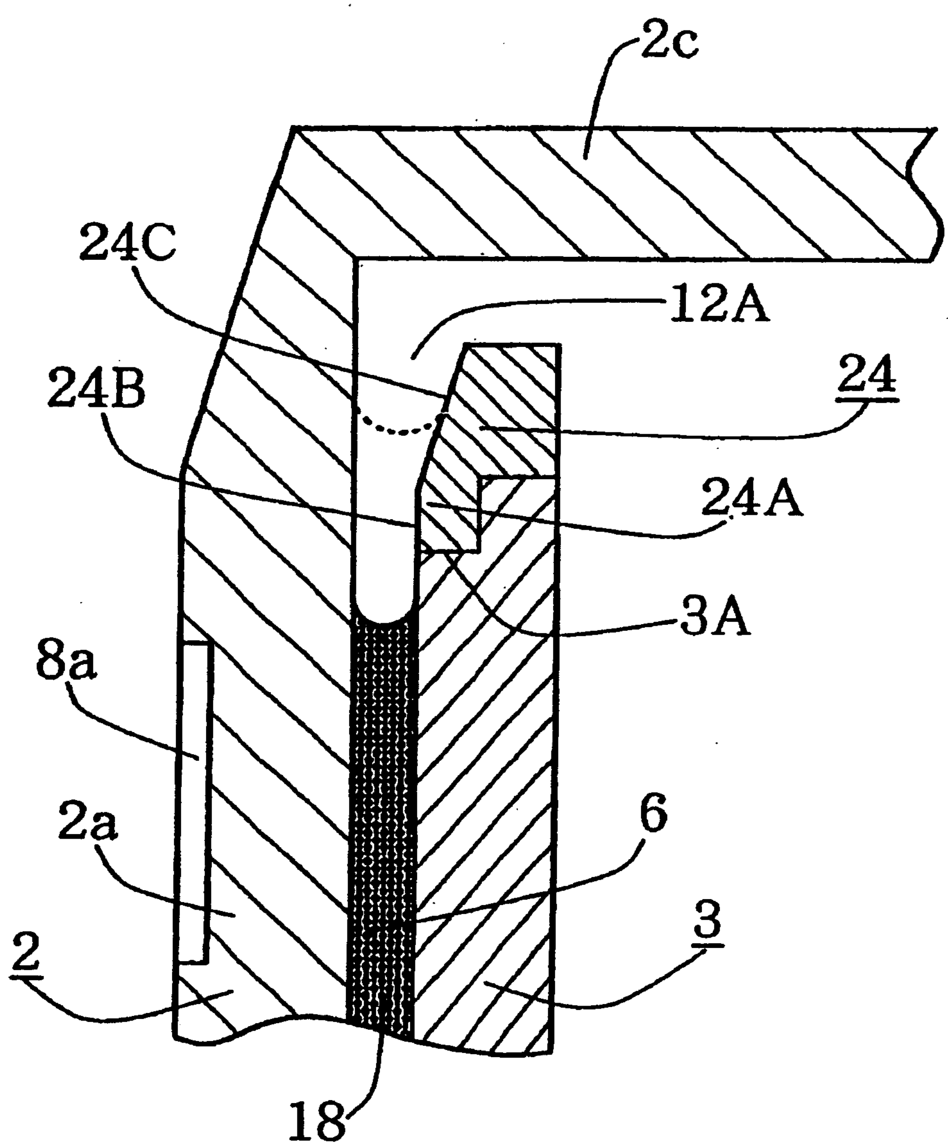


FIG. 10

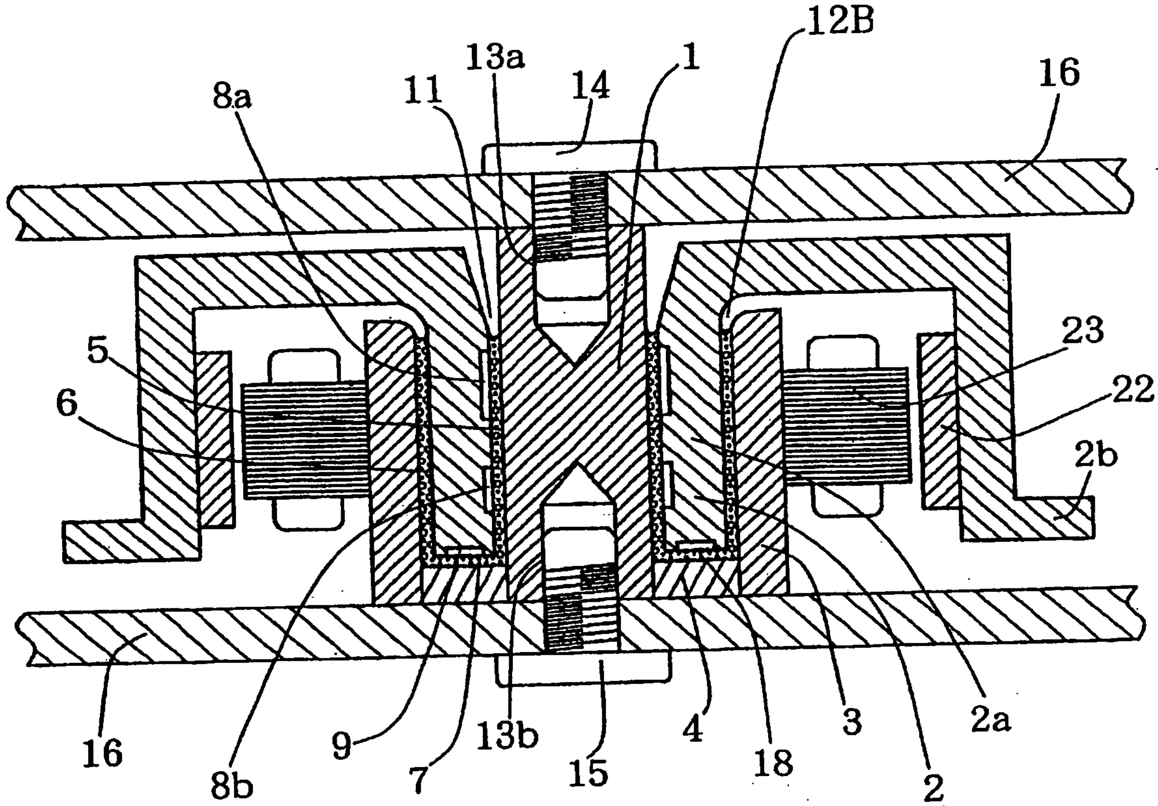


FIG. 11

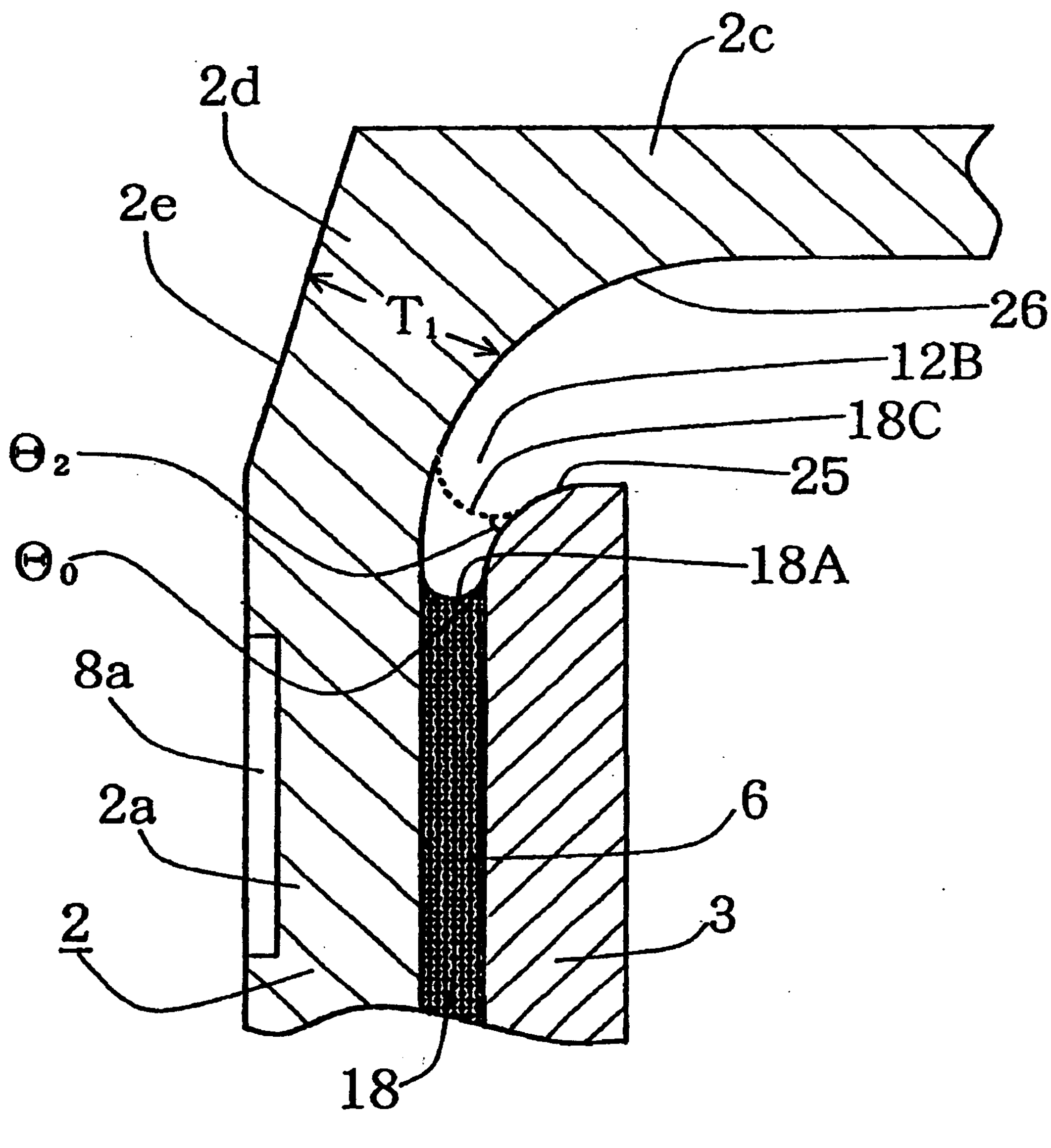


FIG. 12

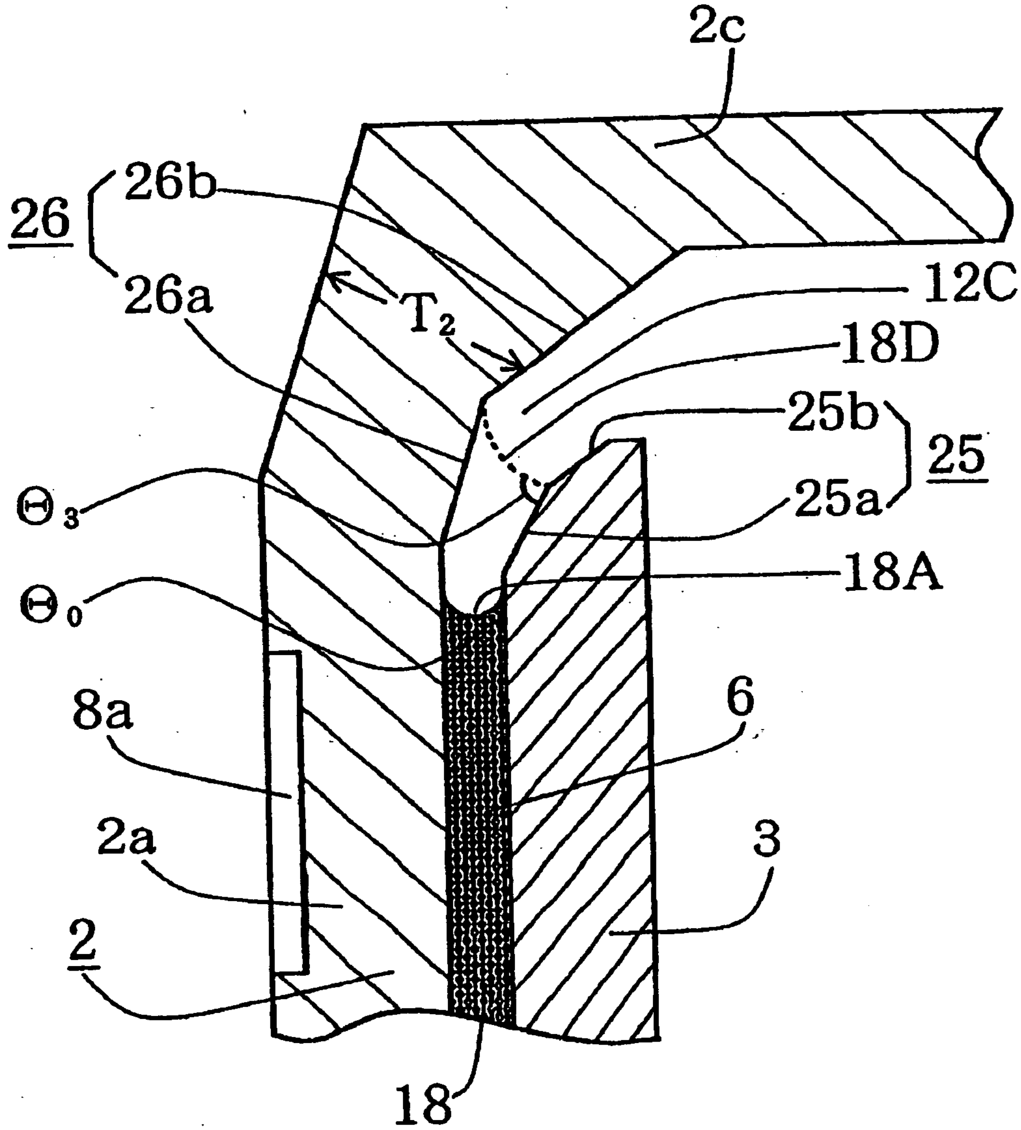


FIG. 13

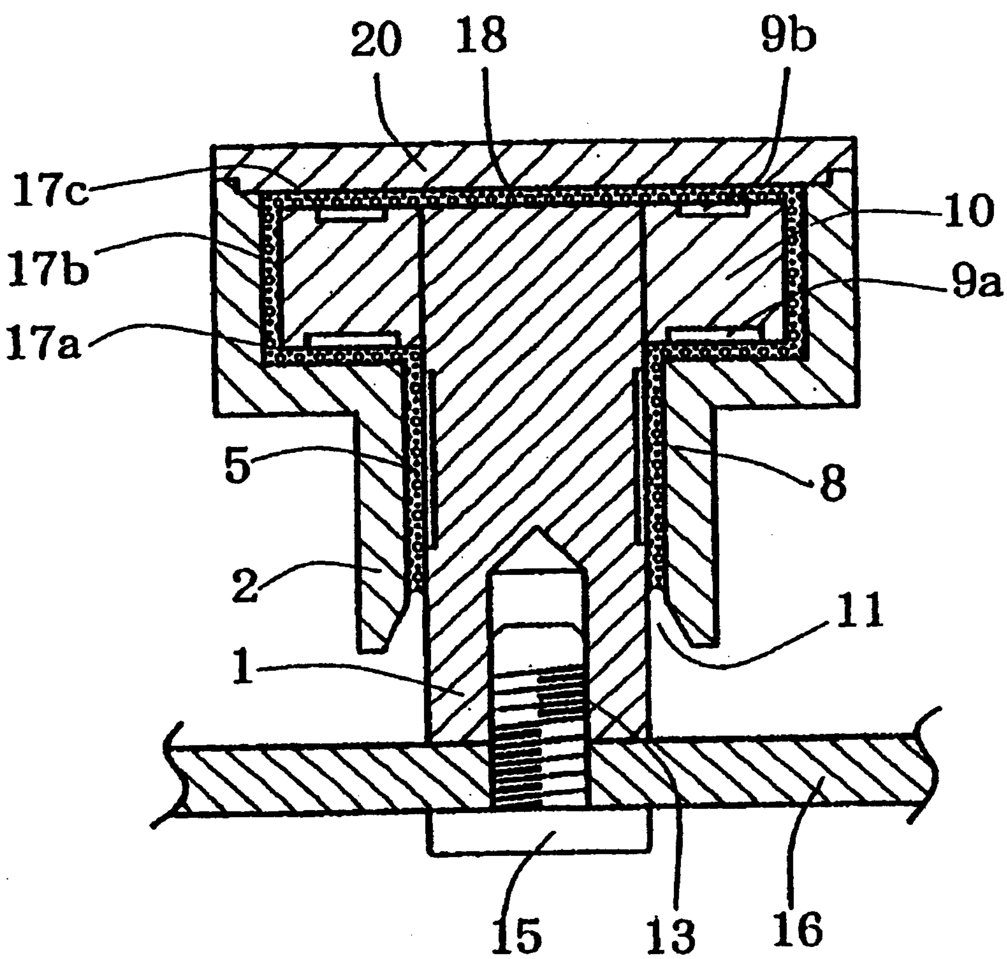
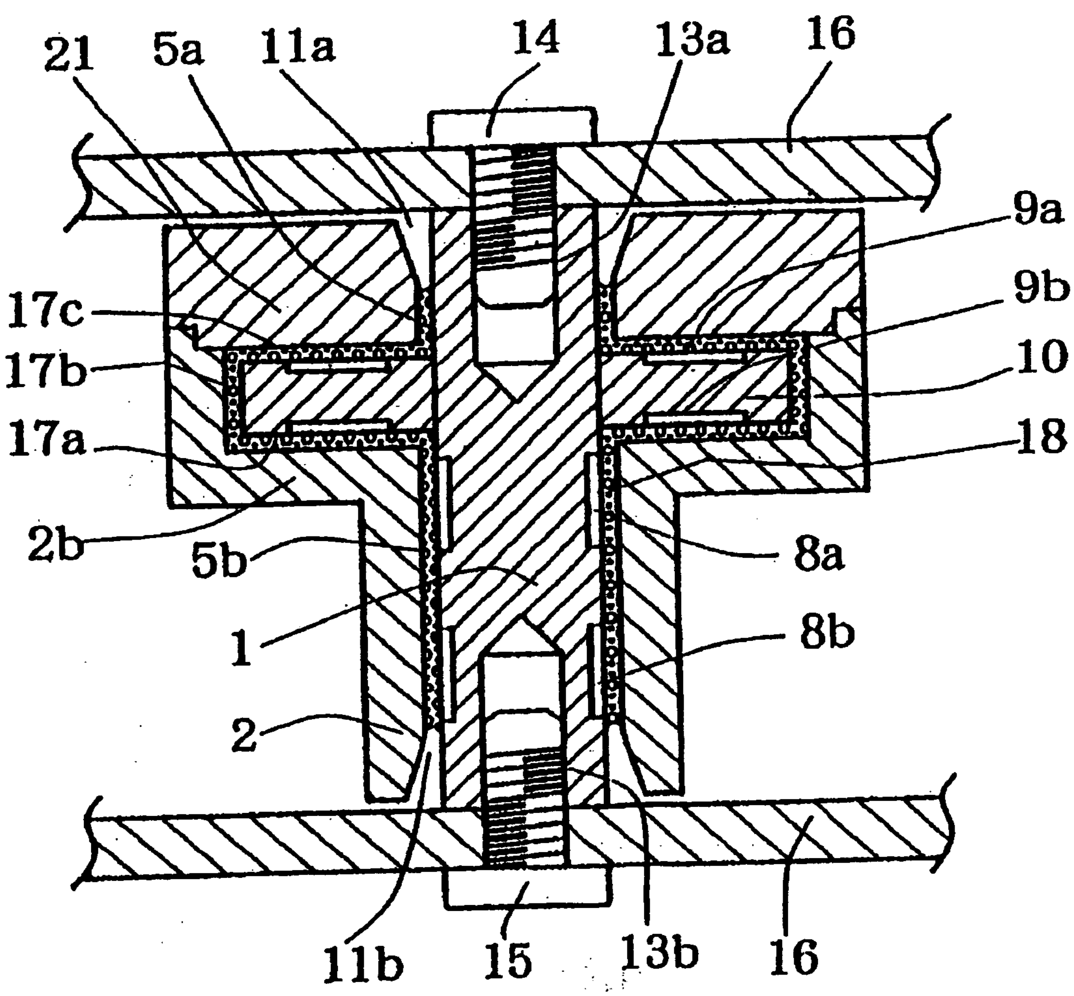


FIG. 14


21
5a
11a
14
13a
16
9a
9b
10
18
17c
17b
17a
2b
5b
8a
1
8b
2
11b
15
13b
16

HYDRAULIC DYNAMIC BEARING AND SPINDLE MOTOR AND ROTARY ASSEMBLY PROVIDED

BACKGROUND OF THE INVENTION

This invention relates to a fluid dynamic pressure bearing adapted as a bearing for rotary apparatuses such as hard disc drive (HDD) units, a spindle motor used as a drive source for such rotary apparatuses and a fluid dynamic pressure bearing adapted for such spindle motors, and more particularly to a both-end fixed-shaft type fluid dynamic pressure bearing having a shaft to be fixed at its respective ends on a chassis, etc. of an apparatus utilized through screwing or the like.

Air dynamic pressure bearings are broadly used in rotary apparatuses such as an HDD, drive optical disc and light polarizing units because of their excellent merits such as their light weight, clean and smooth rotation, durability to heat and cold, long service life and noncontamination to a recording media such as a disc by virtue of not using lubrication oil. In recent years, however, there has been a significant increase in information to be processed. Particularly, are large capacity HDD apparatuses required to rotationally drive as many as five or more disc. This requirement can no longer be met by an air dynamic pressure bearing. In order to cope with this, fluid dynamic pressure bearings have been adopted in HDD apparatuses to support greater load weight than that supported by the air dynamic pressure bearings.

There are disclosures of the basic structure and operation of fluid dynamic pressure bearing , for example, in U.S. Pat. No. 5,112,142; U.S. Pat. No. 5,524,985; U.S. Pat. No. 5,524,986; and U.S. Pat. No. 5,533,812.

The conventional fluid dynamic pressure bearings, particularly fluid dynamic pressure bearings of the sleeve rotation, type include two kinds of devices depending on the ways used to fix the shaft onto an apparatus in which it is utilized. One type is a one-end fixed-shaft type fluid dynamic pressure bearing as shown in FIG. 13, and the other is a both-end fixed-shaft type fluid dynamic pressure bearing as shown in FIG. 14. First, the fluid dynamic pressure bearing of FIG. 13 is structured by a fixed shaft **1** at its lower end fixed on a chassis **16** or the like through a screw **15**, and a rotary sleeve **2** having an upper end completely covered by a lid member **20** and a lower end having an opening **11** forming a capillary seal. Next, the fluid dynamic pressure bearing of FIG. 14 is structured by a fixed shaft **1** fixed at its opposite ends on a chassis **16** or the like of an apparatus utilized through screws **14** and **15**, and a rotary sleeve **2** having openings **11***a* and **11***b* respectively forming upper and lower capillary seals.

In FIG. 13 and FIG. 14, **8**, **8***a*, and **8***b* are radial dynamic pressure producing grooves while **9***a* and **9***b* are thrust dynamic pressure producing grooves. **5**, **5***a*, **5***b*, **17***a*, **17***b* and **17***c* are fine gaps formed between the fixed shaft **1** and the rotary sleeve **2**. These fine gaps are filled therein with lubrication oil **18**. The fine gaps have a width of usually 2 to 15 μm, although depending on the size of the fluid dynamic pressure bearing. **13***a* is an upper screw hole of the fixed shaft, while 13, **13***b* is a lower screw hole.

In the shaft-one-end fixed type fluid dynamic pressure bearing of FIG. 13, the lubrication oil **18** filled within the fine gaps **5**, **17***a*, **17***b* and **17***c* contacts with the air at tapered opening **11**. However, the lubrication oil **18** filled in the gaps is prevented from leaking outside the fine gaps by a capillary seal and surface tension due to the opening **11**. In particular, the fine gaps **17***a*, **17***b* and **17***c* form a closed end.

The filled lubrication oil **18** hardly leaks out through the opening **11** due to a fine gap structure having such a closed end, i.e. a fine gap structure with one-side closure. In the both-end fixed-shaft type fluid dynamic pressure bearing of FIG. **19**, on the other hand, the lubrication oil **18** filled within the fine gaps **5***a*, **5***b*, **17***a*, **17***b* and **17***c* contacts with the air at a tapered upper opening **11***a* and lower opening **11***b*. However, the filled lubrication oil **18** is prevented from leaking out of the fine gaps by the capillary seal and surface tension due to the openings.

Of the above related-art apparatus, the one-end fixed-shaft type fluid dynamic pressure bearing of FIG. **13** has a closed end in the fine gaps. Accordingly, the apparatus, in case tilted, hardly causes the lubrication oil to leak thus being excellent in sealability. However, there is a disadvantage in that the shaft **1** is fixed at only one point of its lower end and undergoes precession motion during rotation at high speed, resulting in instability in rotation. Conversely, the both-end fixed-shaft type dynamic pressure bearing of FIG. **14** fixes the shaft **1** at its both ends and hence does not undergo precession motion, offering stable rotation. However, there is a problem in that the fine gaps are opened to the air at upper and lower sides thus resulting in insufficient sealability. Even if a surface tension is formed by forming an air reservoir in a fine gap between the upper and lower radial dynamic pressure producing grooves **8**a and **8**b, the surface tension abruptly decreases when the fluid dynamic pressure bearing is tilted and positioned in a horizontal direction. Furthermore, when, in this state, temperature change or external impact is applied, the lubrication oil filled within the fine gap readily leaks to the outside.

SUMMARY OF THE INVENTION

It is an object of the present invention to maintain, in a both-end fixed-shaft type fluid dynamic bearing, a high sealability not only during rotation at high speed but also even upon being tilted in a standstill state.

It is another object of the invention to provide a spindle motor which can stably rotate at high speed.

It is still another object of the invention to provide a rotary apparatus which can stably rotationally drive at high speed a rotary member such as a hard disc.

In brief, the present invention is a double sleeve type fluid dynamic pressure bearing, comprising: a fixed shaft having respective ends to be fixed to an apparatus utilized; a rotary sleeve arranged to provide a first fine gap between an inner peripheral surface thereof and an outer peripheral surface of the fixed shaft; a fixed sleeve arranged to provide a second fine gap between an inner peripheral surface thereof and an outer peripheral surface of the rotary sleeve; and wherein the first fine gap and the second fine gap have one ends made as open ends contacting the air while the first fine gap and the second fine gap have the other ends made as closed ends in direct communication with each other, the fine gaps being filled with lubrication oil, and the first fine gap being formed with a dynamic pressure producing groove.

In the double sleeve type dynamic pressure bearing, the second fine gap is greater in width than the first fine gap within a range of capable of producing a dynamic pressure, thereby removing instability during high speed rotation due to a difference in flowing speed of lubrication oil.

In the double sleeve type fluid dynamic pressure bearing, seal means different from a related art capillary seal is provided at the opening of the second fine gap. The seal means is a resinous collar fitted at an outer end of the fixed sleeve. Alternatively, the seal means may be a curved type

annular seal groove formed in the opening of the second fine gap by a first curved wall surface curved radially outward and a second curved wall surface similarly curved radially outward. Furthermore, the seal means may be a multi-staged slant type annular seal groove formed in the opening of the second fine gap by a first plurality slant wall surface having a plurality of annular slant surfaces slanted by stages radially outward and a second plurality slant wall surface having a plurality of annular slant surfaces similarly slanted by stages radially outward.

Also, the present invention is, in a spindle motor structured by a rotor including a rotor magnet, a stator including a stator coil and a fluid dynamic pressure bearing for rotatably supporting the rotor with respect to the stator, the spindle motor adopting for the fluid dynamic pressure bearing a double sleeve type fluid dynamic pressure bearing. The invention is furthermore a rotary apparatus having, as drive source to a rotary member, the spindle motor structured by a rotor including a rotor magnet, a stator including a stator coil and a fluid dynamic pressure bearing for rotatably supporting the rotor with respect to the stator.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view of a spindle motor having a first embodiment of a fluid dynamic pressure bearing according to the present invention;

FIG. 2 is a sectional view of a first modification of a fluid dynamic pressure bearing of the invention;

FIG. 3 is a sectional view of a second modification of a fluid dynamic pressure bearing of the invention;

FIG. 4 is a sectional view of a third modification of a fluid dynamic pressure bearing of the invention;

FIG. 5 is a sectional view of a fourth modification of a fluid dynamic pressure bearing of the invention;

FIG. 6 is a sectional view of a spindle motor having a second embodiment of a fluid dynamic pressure bearing according to the present invention;

FIG. 7 is a sectional view of a spindle motor having a third embodiment of a fluid dynamic pressure bearing according to the present invention;

FIG. 8 is a partially magnified view including a resin-make collar;

FIG. 9 is a partially magnified view including a modification to the resin-make collar;

FIG. 10 is a sectional view of a spindle motor having a fourth embodiment of a fluid dynamic pressure bearing according to the present invention;

FIG. 11 is a partially magnified view including a fluid seal portion;

FIG. 12 is a partially magnified view including a modification to the fluid seal portion;

FIG. 13 is a sectional view of a one-end fixed-shaft type fluid dynamic pressure bearing of a related art; and

FIG. 14 is a sectional view of a both-end fixed-shaft type fluid dynamic pressure bearing of a related art.

BEST MODE FOR PRACTICING THE INVENTION

Referring to FIG. 1, there is shown a sectional view of a double sleeve structure both-end fixed-shaft type dynamic pressure bearing according a first embodiment of the present invention, and a spindle motor having this fluid dynamic pressure bearing. In FIG. 1, the fluid dynamic pressure bearing includes a fixed shaft **1** fixed at respective ends to an apparatus utilized, a rotary sleeve **2** providing a first fine gap **5** cooperatively with the fixed shaft **1**, a fixed sleeve **3** providing a second fine gap **6** cooperatively with the rotary sleeve **2**, and a holder member **4** providing a third fine gap **7** cooperatively with the rotary sleeve **2**.

The fixed shaft **1** is formed with screw holes **13***a* and **13***b* at respective ends. The fixed shaft **1** is firmly fixed to a chassis **16** of an apparatus utilized such as an HDD apparatus through screws **14** and **15** screwed to the screw holes **13***a* and **13***b*. The rotary sleeve **2** is a member formed by a sleeve portion **2***a* having inner and outer peripheral surfaces, a cup-like hub **2***b* for holding a rotary member such as a disc, and a disc formed extended portion **2***c* for firmly fixing the cup-like hub **2***b* at an upper end of the sleeve **2***a*. The disc-formed extended portion **2***c* is a portion in a disc form that is horizontally radially outwardly extended from an upper end of a sleeve portion **2***a* of the rotary sleeve **2**, and formed integral with the sleeve **2***a*. The cup-like hub **2***b* serves also as a rotor member for the spindle motor having a rotor magnet **22** mounted on an inner peripheral surface thereof. The fixed sleeve **3** is a member arranged standing on the base plate of the bearing or spindle motor. In the apparatus of the FIG. **1** embodiment without using a base plate, the fixed sleeve **3** is provided standing adjacent the holder member **4** with its inner peripheral surface fitted liquid-tight to an outer peripheral surface of the disc-like holder member **4** coaxially fixed to the fixed shaft **1**. The fixed sleeve **3** also serves as a stator member for the spindle motor, and has a stator coil **23** mounted on an outer peripheral surface thereof.

A tapered opening **11** is provided at a top end of a first fine gap **5** formed between an outer peripheral surface of the fixed shaft **1** and an inner peripheral surface of the rotary sleeve **2**. Similarly, a tapered opening **12** is also provided at a top end of a second fine gap **6** given between an outer peripheral surface of the rotary sleeve **2** and an inner peripheral surface of the fixed sleeve **3**. A third fine gap **7** is given between an lower end surface of the rotary sleeve **2** and an upper surface of the disc-like holder member **4**, which has one end communicated with a lower end of the first fine gap **5** and the other end communicated with a lower end of the second fine gap **6**. In brief, the third fine gap **7** serves as a closed end with respect to the openings **11** and **12**. Lubrication oil **18** is filled within the first fine gap **5**, second fine gap **6** and third fine gap **7**.

Each of these fine gaps, although exaggeratedly shown in FIG. **1**, is actually a fine gap of a size of approximately 5 to 200 μm. Due to this, the lubrication oil **18** has its liquid levels respectively kept at bottom portions of the tapered openings **11** and **12** by a surface tension and capillary phenomenon, being prevented from leaking to the outside in a usual use state. Moreover, the first fine gap **5** and the second fine gap **6** at their lower ends are communicated through the third fine gap **7**, forming a closed end. Accordingly, even in case the fluid dynamic pressure bearing of the invention is tilted, the lubrication oil **18** filled within these fine gaps hardly leaks to the outside.

Radial dynamic pressure producing grooves **8***a* and **8***b* are provided vertically separated in an inner peripheral surface of the sleeve portion **2***a* of the rotary sleeve **2** forming the first fine gap **5**. A thrust dynamic pressure producing groove **9** is provided in a lower end surface of the sleeve portion **2***a* of the rotary sleeve **2** forming the third fine gap **7**. The radial dynamic pressure producing grooves **8***a* and **8***b* are herringbone grooves but may be of other forms. The thrust dynamic pressure producing groove **9** are herringbone grooves in an annular arrangement but may be of other forms.

The spindle motor thus constructed, when supplied by an energizing current to its stator coil 23, is rotated due to electromagnetic action caused by the current and magnetic field of the rotor magnet 22. Thereupon, a radial dynamic pressure is caused in the first fine gap 5 through the radial dynamic pressure producing grooves 8*a* and 8*b*, and a thrust dynamic pressure is caused in the third fine gap 7 through the thrust dynamic pressure producing groove 9. Thus, the spindle motor smoothly maintains rotation at high speed while supporting a rotary member, such as a hard disc, by these dynamic pressures.

The double sleeve structure both-end fixed-shaft type fluid dynamic pressure bearing of the invention explained above was structured by adopting a circular columnar member for the shaft 1, a cylindrical member for the rotary sleeve 2, and a cylindrical member for the fixed sleeve 3. Now, among various modifications, four embodiments will be described with reference to FIG. 2 to FIG. 5 exaggeratedly showing the fine gaps.

Referring to FIG. 2, a double sleeve structure bothend fixed-shaft type fluid dynamic pressure bearing is shown as a modification for the fluid dynamic pressure bearing of FIG. 1. That is, the double sleeve structure both-end fixed-shaft type fluid dynamic pressure bearing of FIG. 2 is structured by adopting a circular columnar member for a fixed shaft 1, a conical frustum member for a rotary sleeve 2, and a conical-inner-perpheral-surfaced member for a fixed sleeve 3 to cooperate with an conical outer peripheral surface to provide therebetween a second fine gap 6. A first fine gap 5 is given between an outer peripheral surface of the fixed shaft 1 and an inner peripheral surface of the rotary sleeve 2. A third fine gap 7 is given between a bottom surface of the rotary sleeve 2 and a top surface of the holder member 4, to communicate between respective lower ends of the first fine gap 5 and the second fine gap 6 serving as a closed end. The first fine gap 5 and the second fine gap 6 respectively have, at their upper ends, tapered openings 11 and 12 serving as capillary seals for the lubrication oil filled within the fine gaps.

Referring to FIG. 3, a double sleeve structure both-end fixed-shaft type dynamic pressure bearing is shown as a second modification for the fluid dynamic pressure bearing of FIG. 1. That is, the double sleeve structure both-end fixed-shaft type fluid dynamic pressure bearing of FIG. 3 is structured by adopting a circular columnar member for a fixed shaft 1, a member with a semi-spherical portion for a rotary sleeve 2, and a member with an inner peripheral surface providing a second fine gap 6 cooperatively with the semispherical surface of the rotary sleeve 2 for a fixed sleeve 3. The second gap 6 includes a semi-circular gap portion 6*a* and a horizontal gap 6*b*. A first fine gap 5 is given between an outer peripheral surface of the fixed shaft 1 and an inner peripheral surface of the rotary sleeve 2. The first fine gap 5 and the second fine gap 6 at their lower ends are directly communicated with each other thereby forming a closed end. The first fine gap 5 and the second fine gap 6 respectively have, at their upper ends, tapered openings 11 and 12 serving as capillary seals for the lubrication oil filled within the fine gaps.

In the meanwhile, it will be understood from FIG. 1, FIG. 2 and FIG. 3 that a variety of double sleeve structure both-end fixed-shaft type fluid dynamic pressure bearing can be realized depending on a member shape of the rotary sleeve 2 even with the fixed shaft 1 using a circular columnar member. Also, it will be apparent that the rotary sleeve 2 is not limited to the cylindrical member of FIG. 1, the conical frustum member of FIG. 2 or the semi-spherical-portion-having member of FIG. 3 but may adopt various shapes of members including deformed versions of cylindrical, conical frustum and disc members.

Referring to FIG. 4, a double sleeve structure both-end fixed-shaft type dynamic pressure bearing is shown as a third modification for the fluid dynamic pressure bearing of FIG. 1. That is, the double sleeve structure both-end fixed-shaft type fluid dynamic pressure bearing of FIG. 4 is structured by adopting a circular columnar member having upper and lower large diameter portions for a fixed shaft 1, a member having an inner peripheral surface to provide a first fine gap 5 cooperatively with an outer peripheral surface of the fixed shaft 1 for a rotary sleeve 2, and a cylindrical member for a fixed sleeve 3. The first fine gap 5 includes a plurality of vertical gap portions 5a and a plurality of horizontal gap portions 5*b*. A second fine gap 6 is given between an outer peripheral surface of the rotary sleeve 2 and an inner peripheral surface of the fixed sleeve 3. The first fine gap 5 and the second fine gap 6 at their lower ends are communicated with each other to provide a third fine gap 7 serving as a closed end between a bottom surface of the rotary sleeve 2 and a top surface of the holder member 4. The first fine gap 5 and the second fine gap 6 respectively have, at their upper ends, tapered openings 11 and 12 serving as capillary seals for the lubrication oil filled within the fine gaps.

Referring to FIG. 5, a double sleeve structure both-end fixed-shaft type dynamic pressure bearing is shown as a fourth modification for the fluid dynamic pressure bearing of FIG. 1. That is, the double sleeve structure both-end fixed-shaft type fluid dynamic pressure bearing of FIG. 5 is structured by adopting a circular columnar member having at its intermediate portion a small diameter portion for a fixed shaft 1, a member having an inner peripheral surface to provide a first fine gap 5 cooperatively with an outer peripheral surface of the fixed shaft 1 for a rotary sleeve 2, and a cylindrical member for a fixed sleeve 3. The first fine gap 5 includes a plurality of vertical gap portions 5*a* and a plurality of horizontal gap portions 5*b*. A second fine gap 6 is given between an outer peripheral surface of the rotary sleeve 2 and an inner peripheral surface of the fixed sleeve 3. The first fine gap 5 and the second fine gap 6 at their lower ends are communicated with each other to provide a third fine gap 7 serving as a closed end between a bottom surface of the rotary sleeve 2 and a top surface of a holder member 4. The first fine gap 5 and the second fine gap 6 respectively have, at their upper ends, tapered openings 11 and 12 serving as capillary seals for the lubrication oil filled within the fine gaps.

Referring to FIG. 6, there is shown a sectional view of a spindle motor having a second embodiment of a fluid dynamic pressure bearing of the present invention.

The fluid dynamic pressure bearing of FIG. 6 is the same in basic structure as the fluid dynamic pressure bearing of FIG. 1, but different in structure of the second fine gap 6. That is, the fluid dynamic pressure bearing of FIG. 6 has a second fine gap 6 that is wider in gap width than the second fine gap 6 of the fluid dynamic pressure bearing of FIG. 1.

Because the second fine gap 6 is located radially outward of the first fine gap 5, the speed of flowing lubrication oil is not equal between these fine gaps. The speed is greater in the second fine gap 6. It was found that the difference in speed impedes stability during high speed rotation in an actual apparatus and that a high flow speed of the lubrication oil 18 through the second gap 6 increases friction and hence loss.

Accordingly, the second fine gap 6 was made wider than the first fine gap 5 as shown in an exaggerated manner in

FIG. 6. That is, the first fine gap **5** has a width of approximately 5 to 20 μm while the second fine gap **6** has a width of approximately 50 to 500 μm. This removed the instability at high speed rotation due to a difference in speed of lubrication oil. The width of the second fine gap is selected on an empirical basis as best suited depending on the structure and size of the fluid dynamic pressure bearing, lubrication oil property, and so on. Such a size includes a height of the first fine gap **5** and a height of the second fine gap **6**. Incidentally, a width of the third fine gap **7** is selected as approximately same as the first fine gap **5**.

Stable high speed rotation was thus realized in the spindle motor adopting the fluid dynamic pressure bearing that is stable at high speed rotation and low in loss. Accordingly, a large capacity HDD apparatus having this spindle motor as a drive source for a rotation member can rotate a hard disc at high speed smoothly and stably.

Referring to FIG. **7**, there is shown a sectional view of a spindle motor having a third embodiment of a fluid dynamic pressure bearing of the present invention.

The fluid dynamic pressure bearing of FIG. **7** is the same in basic structure as the fluid dynamic pressure bearing of FIG. **1**, but different in fluid seal structure at an opening of the second fine gap **6**. That is, the fluid dynamic pressure bearing of FIG. **1** had, at the opening **12** of the second fine gap **6**, the fluid seal structure made as a capillary seal structure utilizing a slant type annular seal groove having a groove width continuously increasing from its minimum groove portion toward maximum groove width portion, i.e. seal groove formed by the slanted inner peripheral surface of the fixed sleeve **3** extended outward in a continuous fashion and the vertical outer peripheral surface of the rotary sleeve **2**. On the contrary, the fluid dynamic pressure bearing of FIG. **7** has, at an opening **12A** of a second fine gap **6**, a fluid seal structure made in a fluid seal structure utilizing a resin collar **24**. This resin collar **24** is a ring-formed Teflon resin formed member that is nearly same in inner and outer diameters as the fixed sleeve **3**.

The resin collar **24**, as shown in the magnified view of FIG. **8**, is formed at its axially lower side with a fitting annular leg **24A** smaller in outer diameter than its upper side. on the other hand, the fixed sleeve **3** which provides the second fine gap **6** cooperatively with the rotary sleeve **2** is formed at its end with a fitting annular step **3A** having an inner diameter equal to or slightly smaller than the outer diameter of the fitting annular leg **24A** and a length equal to that of the fitting annular leg **24A**. The resin collar **24** at its fitting annular leg **24A** is press-fitted into the fitting annular step **3A** of the fixing sleeve **3**, thereby closely fitting and fixing the resin collar **24** in the end of the fixing groove **3**.

The provision of the resin collar **24** improves wettability of lubrication oil **18** in an area close to the opening **12A** of the second fine gap **6**. Consequently, lubrication oil **18** reaches to a position close to the opening **12A** of the second fine gap **6**. The lubrication oil **18** contacting an inner peripheral surface of the resin collar **24** has an arcuate recess **18B** in a surface smaller in recessing than an arcuate recess **18A** in a surface of a lubrication oil **18** reaching close to the opening **12A** of the second fine gap **6** but not contacting with the inner peripheral surface of the resin collar **24**. In other words, an angle Θ_1 defined between a tangent line of the recess **18B** at a point where the lubrication oil **18** contacts the wall surface of the annular seal groove and the wall surface of the annular seal groove is greater than an angle Θ_0 defined between a tangent line of the recess **18A** at a point where the lubrication oil **18** contacts the vertical wall surface of the fine gap **6** and the vertical wall surface of the fine gap **6**. This is because, where contacting the inner peripheral surface of the resin collar **24** higher in wettability than metal, an increased surface tension acts on the lubrication oil **18** than in a case other than the above. Due to the action of such an increased surface tension, the sealability was improved for the lubrication oil **18** at the opening **12A** of the second fine gap **6**.

Referring to FIG. **9**, there is shown a magnified view of a modification to the resin collar **24**. A resin collar **24** is the same in basic structure as that of FIG. **8** but different in its inner peripheral surface. That is, the resin collar **24** of FIG. **8** had the inner peripheral surface as a cylindrical surface having its the same inner diameter as the inner peripheral surface of the fixed sleeve **3**. On the contrary, the resin collar **24** of FIG. **9** has an inner peripheral surface formed by a lower vertical inner peripheral portion **24B** and a upper stanted inner peripheral surface **24C**. The vertical inner peripheral portion **24B** has a same inner diameter as that of an inner peripheral surface of the fixed sleeve **3**. The slanted inner peripheral portion **24C** is a tapered surface having an inner diameter increasing continuously from a boundary to the vertical inner peripheral surface portion **24B** toward an opening end that is opened to the air.

In FIG. **9**, the resin collar **24** at its fitting annular leg **24A** is press-fitted into the fitting annular step **3A** of the fixing sleeve **3**, thereby closely fitting and fixing the resin collar **24** to an end of the fixed sleeve **3**.

The provision of the fluid seal of FIG. **9** using the resin collar **24** to the opening **12** of the second fine gap **6** further improved the sealability for the lubrication oil **18** in the second fine gap **6**. That is, the sealability for the lubrication oil **18** in the second fine gap **6** was further improved by the seal due to an increased surface tension given by a good wettability of the resin collar **24** and the capillary seal utilizing a slant type annular seal groove formed by the slanted inner peripheral surface portion **24C** and the corresponding vertical outer peripheral surface of the rotary sleeve **2**.

Incidentally, the material of the resin collar **24** may be a polyimide based resin or fluorine based resin, instead of the Teflon based resin. Polyimide based resin exhibits equivalent wettability to that of fluorine based resin, and is excellent in formability.

Referring to FIG. **10**, a sectional view of a spindle motor is shown having a fourth embodiment of a fluid dynamic pressure bearing of the invention.

The fluid dynamic pressure bearing of FIG. **10** is the same in basic structure as the fluid dynamic pressure bearing of FIG. **1**. The different point lies in a structure of an annular seal groove formed in an opening in a second fine gap **6** to have a groove width continuously extending radially outward. That is, in the fluid dynamic pressure bearing of FIG. **1**, the fixed sleeve **3** at its end inner peripheral surface was made as the annular slant inner peripheral surface and further the corresponding outer peripheral surface of the rotary sleeve **2** to the end inner peripheral surface of the fixed sleeve **3** was made in the vertical outer peripheral surface, whereby the slant annular seal groove was formed having a groove width continuously extending radially outward toward the opening **12** of the second fine gap **6**. On the contrary, the fluid dynamic pressure bearing of FIG. **10** adopts a curved-type annular seal groove **12B** different from the above.

That is, the fluid dynamic pressure bearing of FIG. **10** has a fluid seal structure formed by a curved-type annular seal

groove 12B in an opening of a second fine gap 6 and having a groove width continuously extending radially outward, and structured by a first curved wall surface 25 and second curved wall surface 26 both curved radially outward. As shown in FIG. 11 as a partly magnified view of FIG. 10, the first curved wall surface 25 is formed at an end of the fixed sleeve 3 while the second curved wall surface 26 is formed in an opposed outer peripheral surface of the rotary sleeve 2 to the first curved wall surface 25, i.e. in a wall surface connecting a vertical outer peripheral surface of the rotary sleeve 2 and an underside of a horizontal disc-formed extending portion 2c. The first curved wall surface 25 formed in the fixed sleeve 3 and the second curved wall surface 26 formed in the rotary sleeve 2 have respective radii of curvature determined such that the groove width continuously increases from a minimum groove width portion toward a maximum groove width portion of the curved type annular seal groove 12B.

The provision of the curved-type annular seal groove 12B at the opening of the second fine gap 6 improved the lubrication oil sealability at that portion. That is, the lubrication oil 18 reaching the curved-type annular seal groove 12B has an arcuate recess 18C in a surface in smaller in recessing than an arcuate recess 18A in a surface of a lubrication oil 18 reaching close to the opening 12 of the second fine gap 6 but not reaching the curved-type annular seal groove 12B. In other words, an angle Θ_2 defined between a tangent line of the recess 18C at a point where the lubrication oil 18 contacts the curved wall surface of the curved-type annular seal groove 12B and the curved wall surface of the curved-type annular seal groove 12B is greater than an angle Θ_0 a defined between a tangent line of the recess 18A at a point where the lubrication oil 18 contacts the vertical wall surface of the fine gap 6 and the vertical wall surface of the fine gap 6. This is because of increase in surface tension at this portion by curving the pair of wall surfaces of the curved-type annular seal groove 12B with a determined radii of curvature so that the groove width continuously increases from a minimum groove width portion to a maximum groove width portion. Due to the action of such an increased surface tension, the sealability was improved for the lubrication oil 18 at the opening 12 of the second fine gap 6.

Referring to FIG. 12, a multi-staged slant type annular seal groove 12C is shown in a partially magnifying view which is a modification for the curved-type annular seal groove 12B of FIG. 11. The multi-staged slant type annular seal groove 12C has a groove width increasing by stages from its minimum groove width portion toward maximum groove width portion, and formed by a first curved wall surface 25 having a plurality of annular slant surface 25a, 25b slanted by stages adially outward and a second curved surface 26 having a plurality of annular slant surface 26a, 26b also slanted by stages adially outward. The first curved wall surface 25 is formed in an end of a fixed sleeve 3, while the second curved wall surface 26 is formed in an opposed outer peripheral surface of a rotary sleeve 2 to the first curved surface, i.e. in a wall surface of the rotary sleeve 2 connecting a vertical outer peripheral surface and an underside of a horizontal disc-formed extending portion 2c.

The provision of the multi-staged slant type annular seal groove 12C at the opening of the second fine groove 6 improved the lubrication oil sealability at this portion. That is, the lubrication oil 18 reaching the multi-staged slant type annular seal groove 12C has a in arcuate recess 18D in surface in smaller in recessing than in arcuate recess 18A an surface of a lubrication oil 18 reaching close to the opening 12 of the second fine gap 6 but not reaching the multi-staged slant type annular seal groove 12C. In other words, an angle Θ_3 defined between a tangent line of the recess 18D at a point where lubrication oil 18 contacts the annular slant wall surface of the multi-staged slant type annular seal groove 12C and the annular slant wall surface is greater than an angle Θ_0 defined between a tangent line of the recess 18A at a point where the lubrication oil 18 contacts the vertical wall surface of the fine gap 6 and the vertical wall surface. This is because of increase in surface tension at this portion due to the multi-staged slant type annular seal groove 12C having a groove width increasing by stages from a minimum groove width portion to a maximum groove width portion structured by a first plurality slant wall surface having a plurality of annular slant surfaces slanted by stages radially outward and a second plurality slant wall surface having a plurality of annular slant surfaces similarly slanted by stages radially outward. Due to the action of such an increased surface tension, the sealability was improved for the lubrication oil 18 at the opening of the second fine gap 6.

The multi-staged slant type seal groove 12C functions as above and hence constitutes so-called a multi-staged capillary seal. Accordingly, the angle Θ_3 defined between a tangent line of the recess 18D at a point where lubrication oil 18 contacts the annular slant wall surface of the multi-staged slant type annular seal groove 12C and the annular slant wall surface is increased at a boundary to an adjacent capillary seal stage. Thus, the surface tension is increased at this boundary.

In case that the curved-type annular seal groove 12B of FIG. 11 or multi-staged slant type annular seal groove 12C of FIG. 12 is provided in the opening of the second fine gap 6, it is possible to enhance the mechanical strength for a base portion 2d of the horizontal disc-formed extended portion 2c integrally coupling a cup-like hub portion 2b to a sleeve portion 2a of the rotary sleeve 2. For example, in the fluid dynamic pressure bearing, the rotary sleeve 2 at its upper end inner peripheral surface 2e is made as an annular slant surface, in order to provide a capillary seal in the opening 11 of the first fine gap 5, as is clear by reference to FIG. 8. Due to this, the base portion 2d is reduced in thickness T0. Contrary to this, the base portion 2d is enhanced in mechanical strength because greater than the thickness T0 is a thickness T1 of the base portion 2d where a curved type annular seal groove 12B is provided in the opening of the second fine gap 6 and also a thickness T2 of the base portion 2d where a multi-staged slant type annular seal groove 12C is provided in the opening of the second fine gap 6.

Although the present invention was explained hereinabove by way of various embodiments, the scope of the invention should never be limited to these embodiment but be defined by the inventions set forth in the claims attached herewith and those of their equivalencies.

What is claimed is:

1. A fluid dynamic pressure bearing, comprising:

a fixed shaft having at least one of a pair of ends mountable to an apparatus;

a rotary sleeve having an inner peripheral surface arranged adjacent to an outer peripheral surface of the fixed shaft so as to provide a first fine gap therebetween; and

a fixed sleeve having an inner peripheral surface arranged adjacent to an outer peripheral surface of the rotary sleeve so as to provide a second fine gap therebetween;

wherein the first fine gap and the second fine gap each have one open end exposed to air outside the bearing

and one closed end in direct communication with the closed end of the other, the fine gaps are filled with lubrication oil, one of the first and second fine gaps is formed with a dynamic pressure producing groove, and the open ends of the first fine gap and the second fine gap face in the same direction and are filled with the lubrication oil to substantially the same level.

2. A fluid dynamic pressure bearing according to claim 1; wherein the first fine gap and the second fine gap are the same in height at the open ends thereof.

3. A fluid dynamic pressure bearing according to claim 1; wherein the fixed shaft is a circular columnar member, the rotary sleeve is a member surrounding the fixed shaft and having a semi-spherical portion, and the fixed sleeve is a member surrounding the rotary sleeve and having an inner peripheral surface cooperating with a semi-spherical surface of the rotary sleeve to provide said second fine gap therebetween.

4. A fluid dynamic pressure bearing according to claim 1; wherein the fixed shaft has a cylindrical shape.

5. A fluid dynamic pressure bearing according to claim 1; wherein the fixed sleeve has a cylindrical shape, the fixed shaft is disposed centrally within the fixed sleeve, and the rotary sleeve is disposed between the fixed shaft and the fixed sleeve.

6. A fluid dynamic pressure bearing according to claim 1; wherein the first and second fine gaps have the same width.

7. A fluid dynamic pressure bearing according to claim 1; wherein the rotary sleeve has a cylindrical sleeve portion extending between the fixed shaft and the fixed sleeve, a hub portion extending laterally outward from an outer peripheral surface of the cylindrical sleeve portion opposite the fixed shaft, and a disc-shaped portion connecting the cylindrical sleeve portion and the hub portion.

8. A fluid dynamic pressure bearing according to claim 7; wherein the hub portion of the rotary sleeve has a cup-shaped form; and further comprising a magnet disposed on an inner peripheral surface of the hub portion facing the fixed shaft, and a coil disposed on an outer peripheral surface of the fixed sleeve facing the magnet.

9. A fluid dynamic pressure bearing according to claim 1; wherein the fixed shaft has opposing ends each being fixedly mountable to a support member of the apparatus in which the bearing is used.

10. A fluid dynamic pressure bearing according to claim 1; further comprising a third fine gap having a first closed end meeting the closed end of the first fine gap and a second closed end meeting the closed end of the second fine gap.

11. A fluid dynamic pressure bearing according to claim 1; wherein the second fine gap has a greater width than that of the first fine gap and the width of the second fine gap is within a range capable of producing a dynamic pressure.

12. A fluid dynamic pressure bearing according to claim 11; wherein the first fine gap has a width of approximately 5 to 20 μm, and the second fine gap has a width of approximately 50 to 500 μm.

13. A fluid dynamic pressure bearing according to claim 1; wherein the fixed sleeve has a collar formed of a resin serving as a seal disposed at an end thereof.

14. A fluid dynamic pressure bearing according to claim 13; wherein the collar is formed of a fluorine-based resin.

15. A fluid dynamic pressure bearing according to claim 13; wherein the collar is formed of a polyimide-based resin.

16. A fluid dynamic pressure bearing according to claim 1; wherein the rotary sleeve has a disc-shaped extended portion horizontally extending radially outward from an outer end thereof, and an annular seal groove is formed proximate an open end of one of the first fine gap and the second fine gap and comprising a first curved wall surface curved radially outward from a center of the fixed shaft and a second curved wall surface similarly curved radially outward to have a groove width continuously increasing from a minimum groove width portion to a maximum groove width portion.

17. A fluid dynamic pressure bearing according to claim 16; wherein the first curved wall surface is formed at an end of the fixed sleeve, and the second curved wall surface is formed in a wall surface connecting a vertical outer peripheral surface and an underside of the disc-shaped extended portion of the rotary sleeve.

18. A fluid dynamic pressure bearing according to claim 1; wherein the rotary sleeve has a disc-shaped extended portion horizontally extending radially outward from an outer end thereof, and a multi-staged slant type annular seal groove is formed in an open end of the second fine gap and comprises a first slant wall surface having a plurality of annular slant surfaces slanted in stages radially outward with respect to the fixed shaft and a second slant wall surface having a plurality of annular slant surfaces similarly slanted in stages radially outward and curved radially outward so as to have a groove width increasing in stages from a minimum groove width portion to a maximum groove width portion.

19. A fluid dynamic pressure bearing according to claim 18; wherein a first slant wall surface is formed at an end of the fixed sleeve, and the second slant wall surface is formed in a wall surface connecting a vertical outer peripheral surface and an underside of the horizontally extended portion of the rotary sleeve.

20. A fluid dynamic pressure bearing according to claim 1; wherein each of the first fine gap and the second fine gap are substantially the same in height at the open ends thereof.

21. A fluid dynamic pressure bearing according to claim 1; wherein the open end of the first fine gap has a tapered opening formed between the outer peripheral surface of the fixed shaft and the inner peripheral surface of the rotary sleeve, and the open end of the second fine gap has a tapered opening formed between the outer peripheral surface of the rotary sleeve and the inner peripheral surface of the fixed sleeve.

22. A fluid dynamic pressure bearing, comprising:

a fixed shaft having at least one of a pair of ends mountable to an apparatus;

a rotary sleeve having an inner peripheral surface arranged adjacent to an outer peripheral surface of the fixed shaft so as to provide a first fine gap therebetween;

a fixed sleeve having an inner peripheral surface arranged adjacent to an outer peripheral surface of the rotary sleeve so as to provide a second fine gap therebetween; and

a holding member for holding the fixed shaft and the fixed sleeve so as to form a liquid-tight opening at one end of the fixed sleeve and cooperating with a lower end of the rotary sleeve to provide a third fine gap therebetween;

wherein the first fine gap and the second fine gap each have one open end exposed to air outside the bearing and one closed end in communication with the closed end of the other through the third fine gap, the fine gaps are filled with lubrication oil, one of the first and the second fine gaps is formed with a dynamic pressure producing groove, and the open ends of the first fine gap and the second fine gap face in the same direction and are filled with the lubrication oil to substantially the same level.

23. A fluid dynamic pressure bearing according to claim 22; wherein the fixed shaft is a circular columnar member, the rotary sleeve is a cylindrical member surrounding the fixed shaft, and the fixed sleeve is a cylindrical member surrounding the rotary sleeve.

24. A fluid dynamic pressure bearing according to claim 22; wherein the fixed shaft is a circular columnar member, the rotary sleeve is a conical frustum member surrounding the fixed shaft, and the fixed sleeve is a member surrounding the rotary sleeve and having an inner peripheral surface cooperating with a conical surface of the rotary sleeve to provide the second fine gap therebetween.

25. A fluid dynamic pressure bearing according to claim 22; wherein the fixed shaft is a circular columnar member having a central portion having a given diameter and upper and lower Portions surrounding the central portion and having a larger diameter than the central portion, the central portion having a larger length in an axial direction of the fixed shaft than the upper and lower portions, and the rotary sleeve is a member surrounding the fixed shaft and having an inner peripheral surface cooperating with an outer peripheral surface of the fixed shaft to form the first fine gap therebetween.

26. A fluid dynamic pressure bearing according to claim 22; wherein the fixed shaft is a circular columnar member having a central portion provided with a given diameter at an intermediate position of the fixed shaft and upper and lower portions surrounding the central portion and having a larger diameter than the central portion, the central portion having a smaller length in an axial direction of the fixed shaft than the upper and lower portions, and the rotary sleeve is a member surrounding the fixed shaft and having an inner peripheral surface cooperating with an outer peripheral surface of the fixed shaft to form the first fine gap.

27. A fluid dynamic pressure bearing according to claim 22; wherein the second fine gap has a greater width than that of the first fine gap and the width of the second fine gap is within a range capable of producing a dynamic pressure.

28. A fluid dynamic pressure bearing according to claim 27; wherein the first fine gap has a width of approximately 5 to 20 μm, and the second fine gap has a width of approximately 50 to 500 μm.

29. A fluid dynamic pressure bearing according to claim 22; wherein the fixed sleeve has a collar formed of a resin serving as a seal disposed at an end thereof.

30. A fluid dynamic pressure bearing according to claim 29; wherein the collar is formed of a fluorine-based resin.

31. A fluid dynamic pressure bearing according to claim 29; wherein the collar is formed of a polyimide-based resin.

32. A fluid dynamic pressure bearing according to claim 22; wherein the rotary sleeve has a disc-shaped extended portion horizontally extending radially outward from an outer end thereof, and an annular seal groove is formed proximate an open end of one of the first fine gap and the second fine gap and comprising a first curved wall surface curved radially outward from a center of the fixed shaft and a second curved wall surface similarly curved radially outward to have a groove width continuously increasing from a minimum groove width portion to a maximum groove width portion.

33. A fluid dynamic pressure bearing according to claim 32; wherein the first curved wall surface is formed at an end of the fixed sleeve, and the second curved wall surface is formed in a wall surface connecting a vertical outer peripheral surface and an underside of the disc-shaped extended portion of the rotary sleeve.

34. A fluid dynamic pressure bearing according to claim 22; wherein the rotary sleeve has a disc-shaped extended portion horizontally extending radially outward from an outer end thereof, and a multi-staged slant type annular seal groove is formed in an open end of the second fine gap and comprises a first slant wall surface having a plurality of annular slant surfaces slanted in stages radially outward with respect to the fixed shaft and a second slant wall surface having a plurality of annular slant surfaces similarly slanted in stages radially outward and curved radially outward so as to have a groove width increasing in stages from a minimum groove width portion to a maximum groove width portion.

35. A fluid dynamic pressure bearing according to claim 34; wherein a first slant wall surface is formed at an end of the fixed sleeve, and the second slant wall surface is formed in a wall surface connecting a vertical outer peripheral surface and an underside of the horizontally extended portion of the rotary sleeve.

36. A fluid dynamic pressure bearing according to claim 22; wherein each of the first fine gap and the second fine gap are substantially the same in height at the open ends thereof.

37. A fluid dynamic pressure bearing according to claim 22; wherein the open end of the first fine gap has a tapered opening formed between the outer peripheral surface of the fixed shaft and the inner peripheral surface of the rotary sleeve, and the open end of the second fine gap has a tapered opening formed between the outer peripheral surface of the rotary sleeve and the inner peripheral surface of the fixed sleeve.

38. In a spindle motor comprising a rotor including a rotor magnet, a stator including a stator coil and a bearing for rotatably supporting the rotor with respect to the stator so that a current in the coil causes the rotor to undergo rotation; wherein the bearing comprises the fluid dynamic pressure bearing according to either claim 1 or claim 22.

39. A rotary apparatus having a rotationally driven member and a drive source for driving the driven member; wherein the drive source comprises the spindle motor according to claim 38.

40. A bearing comprising: a fixed shaft having at least one end fixedly mountable to an apparatus; a rotary sleeve arranged coaxially with the fixed shaft so that a first fine gap is formed between an inner peripheral surface of the rotary sleeve and an outer peripheral surface of the fixed shaft; a fixed sleeve arranged coaxially with the rotary sleeve so that a second fine gap is formed between an inner peripheral surface of the fixed sleeve and an outer peripheral surface of the rotary sleeve; and a lubrication oil filled in the fine gaps; wherein the first fine gap and the second fine gap each have an open end exposed to air outside the bearing, each of the open ends facing in the same direction, and an opposite end that is not exposed to the air, each of the opposite ends of the first and second fine gaps meet each other, and the first and second fine gaps being filled with the lubrication oil to substantially the same level.

41. A bearing according to claim 40; further comprising a holding member for holding the fixed shaft and the fixed sleeve disposed adjacent to a lower end surface of the rotary

sleeve to provide a third fine gap between the holding member and the lower end surface of the rotary sleeve.

42. A bearing according to claim **41**; wherein the opposite ends of the first and second fine gaps meet each other through the third fine gap.

43. A bearing according to claim **41**; wherein the third fine gap is formed with a thrust dynamic pressure producing groove.

44. A bearing according to claim **40**; wherein a peripheral surface of at least one of the fixed shaft, the rotary sleeve and the fixed sleeve forming at least one of the first and second fine gaps has a dynamic pressure producing groove formed therein.

45. A fluid dynamic pressure bearing according to claim **40**; wherein the open end of the first fine gap has a tapered opening formed between the outer peripheral surface of the fixed shaft and the inner peripheral surface of the rotary sleeve, and the open end of the second fine gap has a tapered opening formed between the outer peripheral surface of the rotary sleeve and the inner peripheral surface of the fixed sleeve.

46. A bearing according to claim **40**; wherein each of the first fine gap and the second fine gap are substantially the same in height at the open ends thereof.

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